JULY 1934

MACHINE DESIGN

AS IT AFFECTS

ENGINEERING—PRODUCTION—SALES

MACHINE DESIGN

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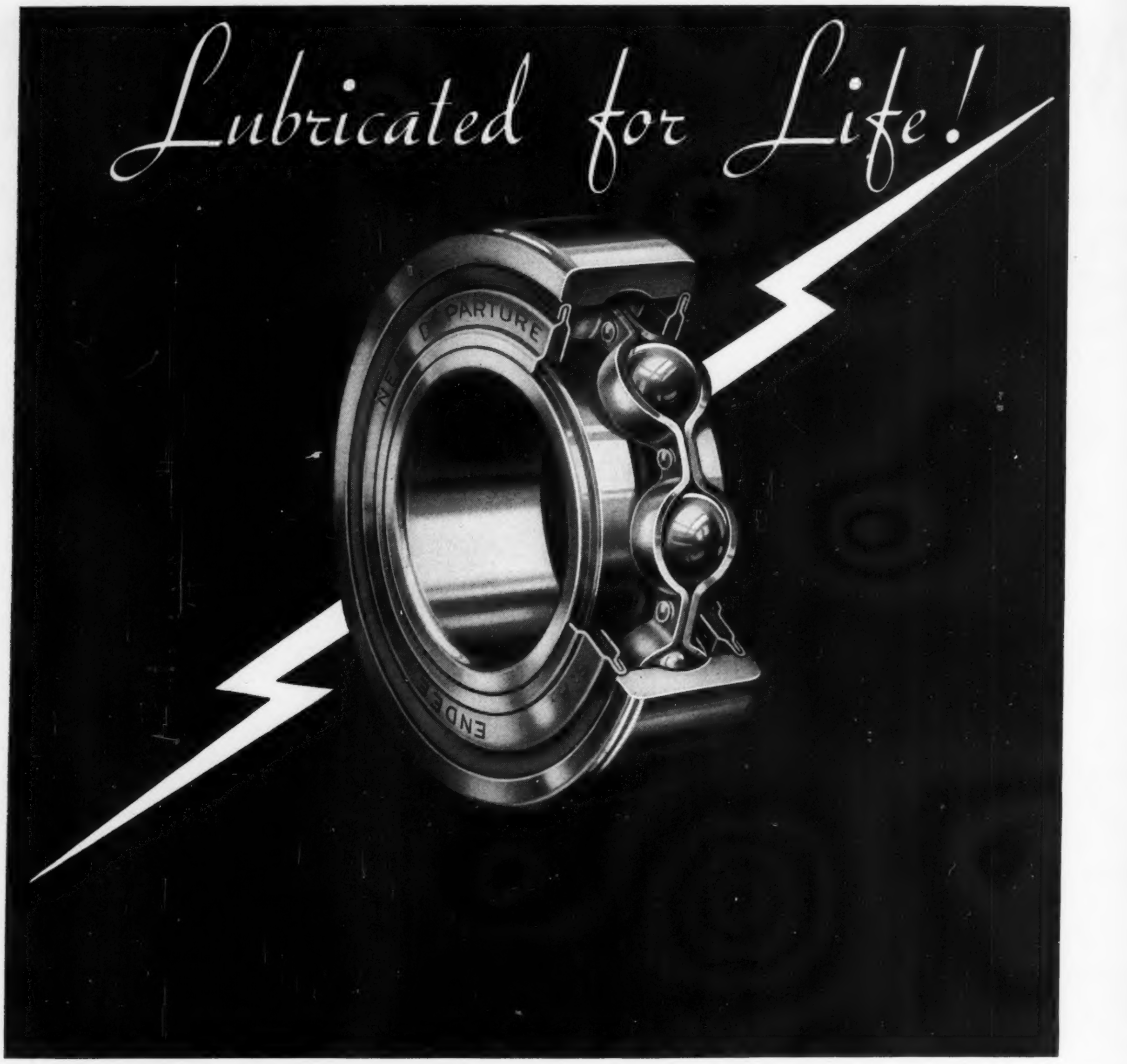

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ITEMIZED INDEX

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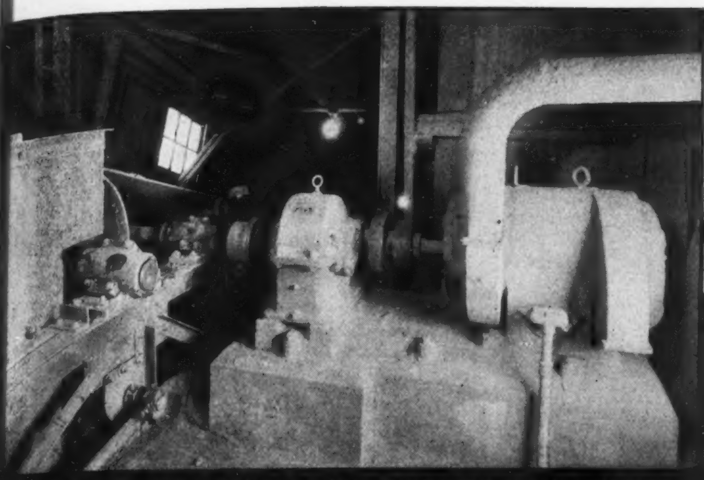
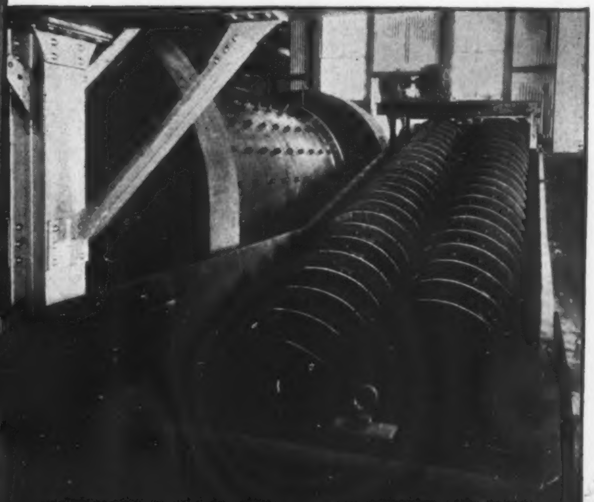
Key: Edit., Editoral Pages; Adv., Advertising Pages; R, Right hand column; L, Left hand column



NO FORCED SHUT-DOWNS ON THIS CONVEYOR DRIVE

★ Assurance of constant efficiency is made certain by this D. O. James Speed Reducer installation on a Rotary Classifier.



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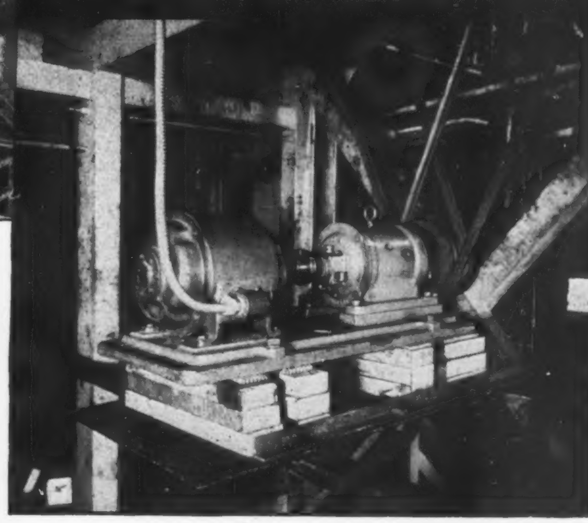
he correct type and ze of Speed Reducer is important a factor in ficient power applica- on as the motor itself. he D. O. James line is mplete to meet every ssible requirement.

close-up view of e crusher drive Noranda Mines.



★ The ability of D. O. James Speed Reducers to withstand the punishment of continuous heavy service justifies their selection for this twin-drive installation.

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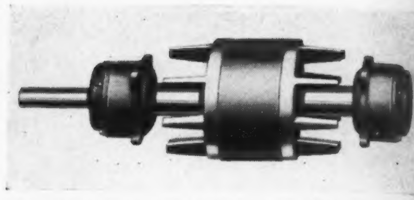
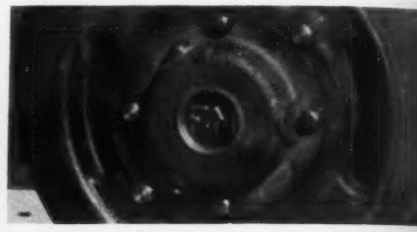
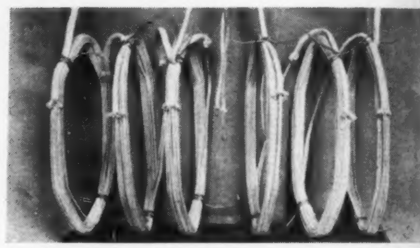
Pioneers in motor building progress, Fairbanks-Morse asks only an investigation of how much *more* these motors have to offer. Start your investigation by writing for full information. Address Fairbanks, Morse & Co., 900 S. Wabash Avenue, Chicago, Ill.



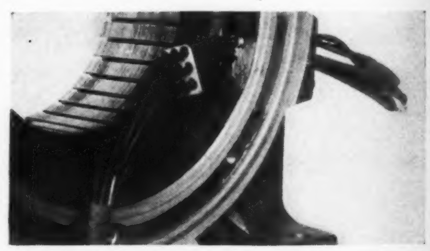
Complete rotor assembly with cartridge-type sealed ball bearings. Note rotor winding is of one-piece construction.



Lubricate sealed ball bearings once a year with tube contained lubricant. Bearings, dust tight. No lubrication drip.



Group wound coils—an entire phase group in a single piece of wire—lead connections from each group *welded*, not soldered or brazed.



Sealed-in leads through frame opening—anchored permanently. No chance for strain on field leads.

Slot insulation — self locking by means of cuff construction — permanent and additional protection for field windings.

Final vibrometer test—one of a series to insure a smooth running motor with minimum vibration.

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MOTORS

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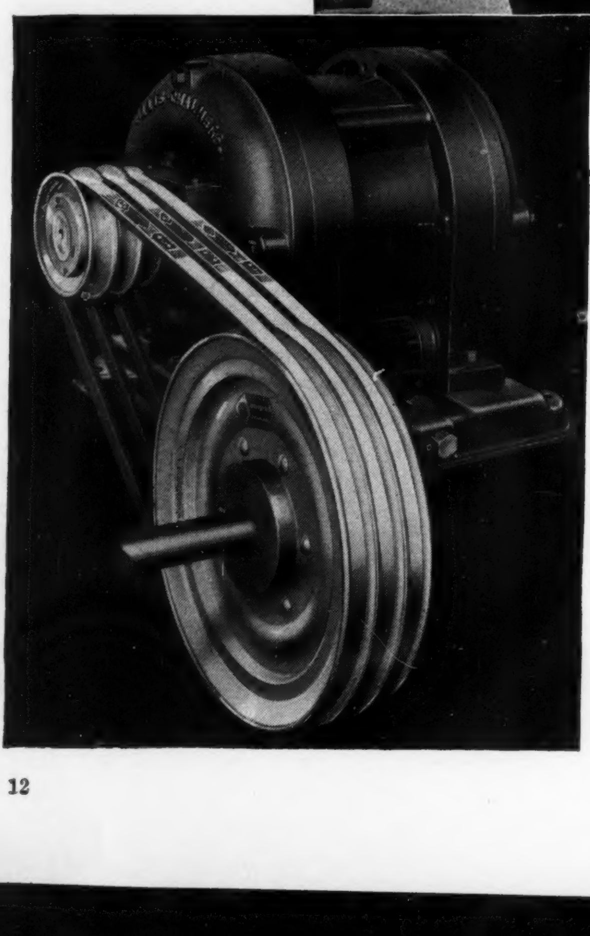
Built for Dependability and Economy.....

Texrope Belts, made for Allis-Chalmers by B. F. Goodrich, are formed in precisely machined molds. They seat perfectly in the grooves of Texrope Sheaves.



SERVICE records demonstrate the ability of Texsteel Sheaves to withstand stress and give maximum performance. Texsteel Sheaves are accurately formed of tough steel and strongly welded at the rim and web to resist shock and vibration. They are truly rigid in construction—light in weight and accurately balanced to assure maximum efficiency. They are attractively finished in a high grade aluminum lacquer which provides for permanent protection.

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MILWAUKEE, WISCONSIN

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TEXROPE DRIVES
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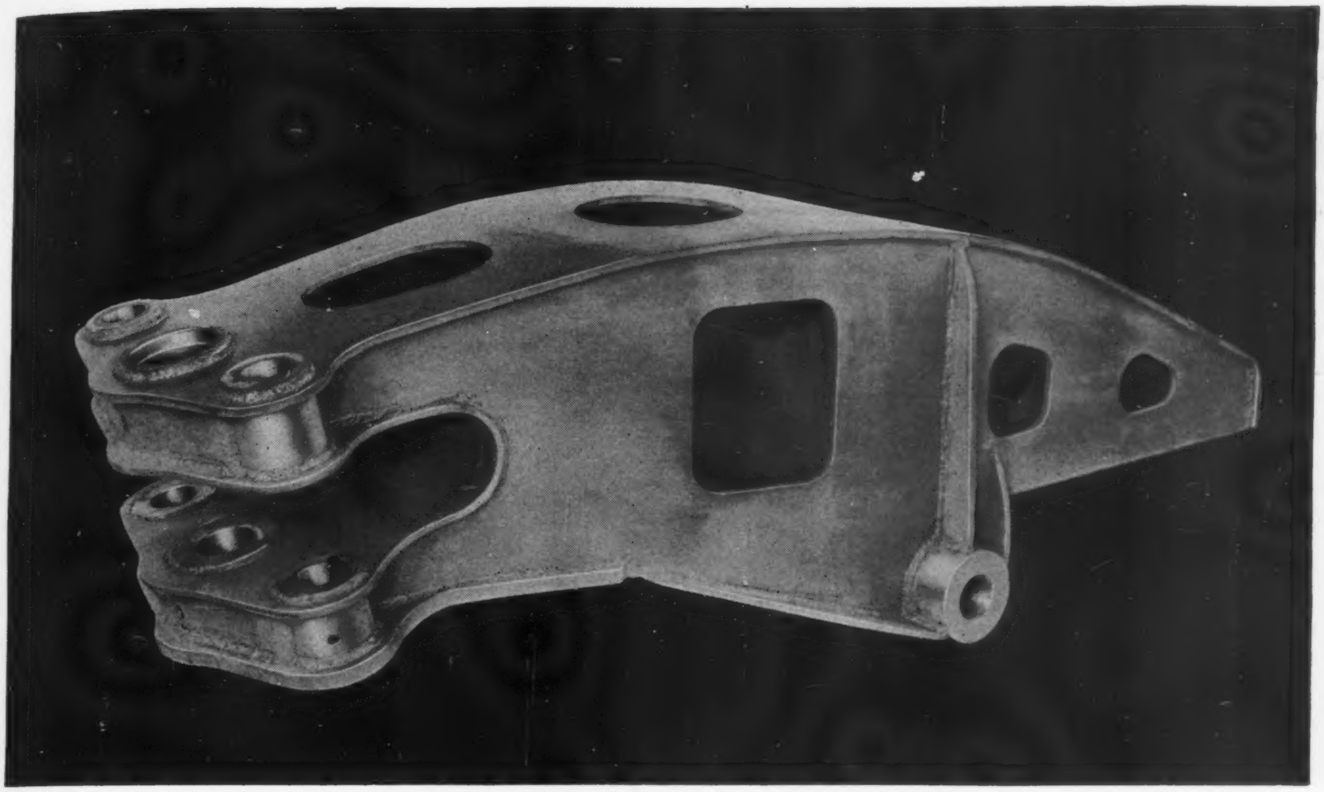

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you can get it with ROLLED STEEL

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5. **Inventories can be kept down . . . pattern storage is reduced.**
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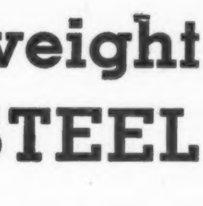
7. **Permits adaptation of standard machines to unusual requirements.**
8. **Modernizes appearance—flat, clean surfaces and straight lines are easily obtained with rolled steel.**

Rolled steel is modernizing machine design. Here's a typical case: a fairlead on a large dragline excavator. As now made of welded steel plates it provides maximum strength and rigidity with a minimum of materials. There isn't a pound of waste metal in it.

Rolled steel has the highest resistance to deformation of any material available for practical machine construction.

The Illinois Shape Book belongs in every modern engineering or design department. Write today for a copy of this 280-page book containing profiles and physical data on a wide variety of rolled steel products.



US STEEL

Illinois Steel Company

208 SOUTH LASALLE STREET, CHICAGO, ILL.
SUBSIDIARY OF UNITED STATES STEEL CORPORATION

Calendar of Meetings and Expositions

Aug. 26-30—

Leipzig Trade Fair.

Fall fair to be held at Leipzig, Germany. Information on the exposition may be obtained from Leipzig Trade Fair Inc., 10 East Fortieth street, New York.

Aug. 27-Sept. 1—

National Association of Power Engineers.

Exposition and annual meeting to be held at Curtis hotel, Minneapolis. Fred W. Raven, 1140 Lake street, Oak Park, Ill., is secretary of the association.

Sept. 3-7—

American Institute of Electrical Engineers.

Annual Pacific Coast meeting to be held at Salt Lake City, Utah. H. H. Henline, 33 West Thirty-ninth street, New York, is secretary of the institute.

Sept. 10—

Technical Association of the Pulp and Paper Industry.

Semiannual meeting to be held at Multnomah hotel, Portland, Ore. R. G. Macdonald, 370 Lexington avenue, New York, is secretary of the association.

Sept. 10-14—

American Chemical society.

Semiannual meeting to be held at Cleveland. Dr. Charles L. Parsons, 728 Mills building, Washington, is secretary of the society.

Sept. 18-20—

Association of Iron and Steel Electrical Engineers.

Exposition of equipment and annual meeting to be held at Hotel Statler, Cleveland. John F. Kelly, 1010 Empire building, Pittsburgh, is managing director of the association.

Sept. 24-27—

American Transit association.

Exposition of equipment and annual meeting to be held in the Public Auditorium, Cleveland. Guy C. Hecker, 292 Madison avenue, New York, is secretary of the association.

Oct. 1-5—

American Society for Metals.

Annual meeting and exposition to be at Commerce hall, Port of Authority building, New York. W. H. Eisenman, 7016 Euclid avenue, Cleveland, is secretary of the society.

Oct. 1-5—

National Safety council.

Exposition and annual meeting to be held at the Cleveland, Statler, Carter and Hollenden hotels, Cleveland. W. H. Cameron, 20 North Wacker drive, Chicago, is man-managing director of the council.

Oct. 1-5—

American Welding society.

Annual meeting to be held at New York. M. M. Kelly, 33 West Thirty-ninth street, New York, is secretary of the society.

Oct. 8-13—

Brewery Supply and Equipment Exposition.

To be held at Grand Central Palace, New York. Information on the exposition may be obtained from Charles F. Roth, Grand Central Palace, New York.

Oct. 15-20—

Dairy and Ice Cream Machinery and Supplies association.

Exhibition of equipment and annual meeting to be held at Public Auditorium, Cleveland. C. E. Breece, 232 Madison avenue, New York, is secretary of the association.

Oct. 22-26—

American Foundrymen's association.

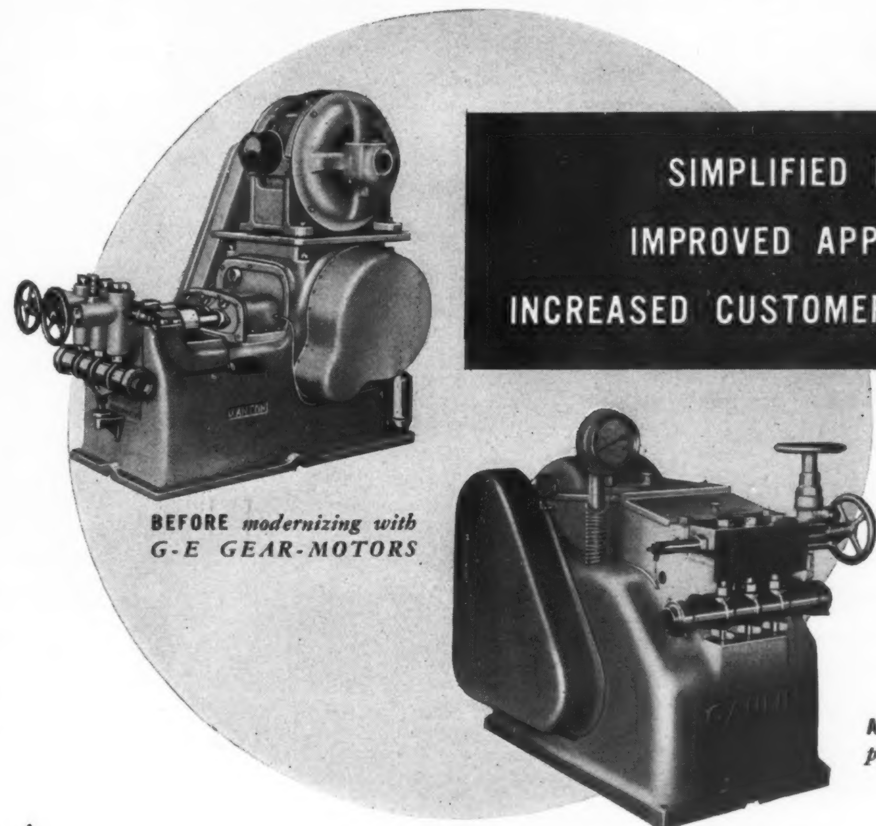
Annual meeting, exposition and International Foundry congress to be held at the Auditorium, Philadelphia. C. E. Hoyt, 222 West Adams street, Chicago, is secretary of the association.

Oct. 29-Nov. 2—

American Gas association

Exposition and annual meeting to be held at the Auditorium, Atlantic City, N. J. K. R. Boyes, 420 Lexington avenue, New York, is secretary of the association.

HOW G-E GEAR-MOTORS HELPED MANTON-GAULIN



SIMPLIFIED DESIGN

IMPROVED APPEARANCE

INCREASED CUSTOMER SATISFACTION

BEFORE *modernizing with G-E GEAR-MOTORS*

AFTER—*trim, compact, pleasing to the eye*

A GLANCE at the "before" type and the modernized "after" tells what happened. Down from the top came the standard motor; into the base went a G-E GEAR-MOTOR. This resulted in simplified design.

With the motor tucked out of sight, the possibility of making other desirable refinements was quickly evident and readily effected—"streamlined" styling was made relatively easy.

The net result for this large machinery manufacturer: a more compact machine, still further reduction of maintenance, and, to quote Mr. J. A. Clark, Production Manager: "We have repeatedly been advised by our customers that they are very much pleased with the changes."

Like Manton-Gaulin, you may find it profitable when planning a new machine, or redesigning an old one, to give thought to the contributions G-E gear-motors can make to the perfection of your ideas through reduction in size and weight, increase in flexibility and efficiency, and the sales appeal of bettered appearance.

We should be pleased to send you a copy of bulletin GEA-1437A, which discusses G-E gear-motors in detail; specialists in machine electrification will gladly call and help you with your electrical problems. Address the nearest G-E office, or General Electric, Schenectady, N. Y.

Your requirements for low speed in any rating from $\frac{1}{6}$ to 75 hp. can easily and quickly be met with **G-E GEAR-MOTORS**

020-50

GENERAL ELECTRIC

Design for OXWELDING

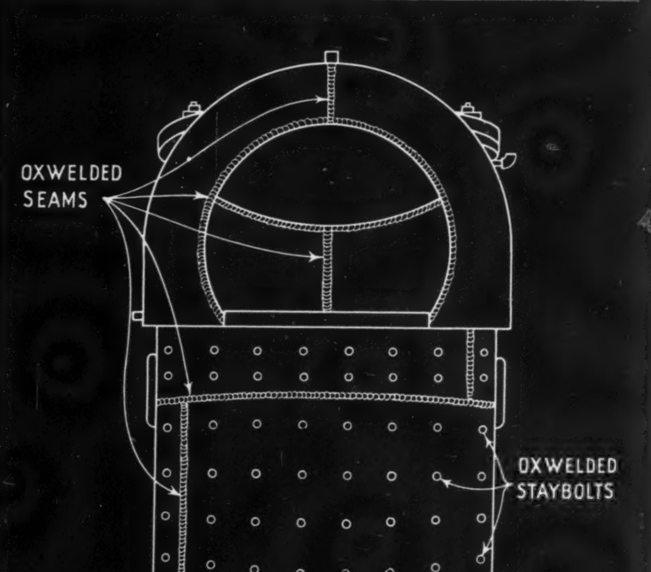
get Strength with Simplicity

A PAPER manufacturer needed a number of alkali recovery furnaces built for several mills. Unsatisfied with previous designs because of inherent weaknesses requiring bulky reinforcement of the joints, the chief engineer called on Linde Process Service to help redesign for oxwelding.

The oxwelded butt joint, which Linde recommended, was adopted for the new design and specifications were prepared for oxy-acetylene welding with High Test steel welding rod. This joint, by developing the full strength of the base material, permitted design changes which made the furnace more efficient and easier to operate and maintain.

Design for oxwelding also gave greater strength and heat resistance, less weight and bulk, more rigidity, and simpler and speedier fabrication.

Does your product need these features? If so, the trained Linde representative, who will call at your request, may be able to assist you in its design or redesign. The nearest Linde Sales Office will be glad to explain this service to you, and to suggest where you can utilize the oxy-acetylene process profitably.



Linde Process Service helped to make this a better furnace. With strong, oxwelded joints its fabrication was simplified. New design features such as the flat top and oxwelded staybolts to make the double wall shell rigid were incorporated without increasing the cost.

THE LINDE AIR PRODUCTS COMPANY

Unit of Union Carbide and Carbon Corporation

126 Producing Plants UCC 627 Warehouse Stocks

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NRA

MACHINE DESIGN
THE JOHNSON PUBLISHING CO., CLEVELAND, OHIO
July, 1934 Vol. 6—No. 7

Shrinking Cloth Mechanically in Modernized Machine

By C. H. Ramsey
President, Morrison Machine Co.

ANY man who has been harassed by a shirt collar uncomfortably tight, through laundry shrinkage, will be among the first to defend the statement that the Sanforizing process is one of the outstanding contributions to the art of textiles in the present century.

Practically all textiles are woven under tension in both warp and filling. For obvious reasons, textile machinery is designed to operate this way. From the moment the gray goods enter upon their course of dyeing, bleaching and finishing, they are subjected to a continuous repetition of pulling and stretching, more particularly warpwise, until they are ready to be delivered to the cutting up trade. When fabric thus treated is finally made up into garments, it awaits only a favorable opportunity to change its dimensions. This opportunity occurs as soon as the garments are dampened or immersed in water. The water acts as a lubricant and allows the fibers to readjust themselves. The fibers also swell and, as the yarns are twisted, this swelling causes a shortening thereof. The combination of swelling and shortening, owing to the twist, further causes a shrinkage of the fabric because of this readjustment of the position of the yarn.

Inasmuch as the principal cause of the shrinkage in fabrics when they are subjected to laundering is mechanical, the most effective treatment to prevent shrinking may be found in mechanically re-arranging the fibers to the same extent that they would arrange themselves when subjected to a full washing in a laundry.

Sanforizing is, broadly stated, a process mechanically effecting any desired combination of dimensional change in fabric, at the same time conserving and usually improving its finish.

The complete machinery used in the Sanforizing process consists of a feeding machine, sometimes referred to as a hold-back machine.

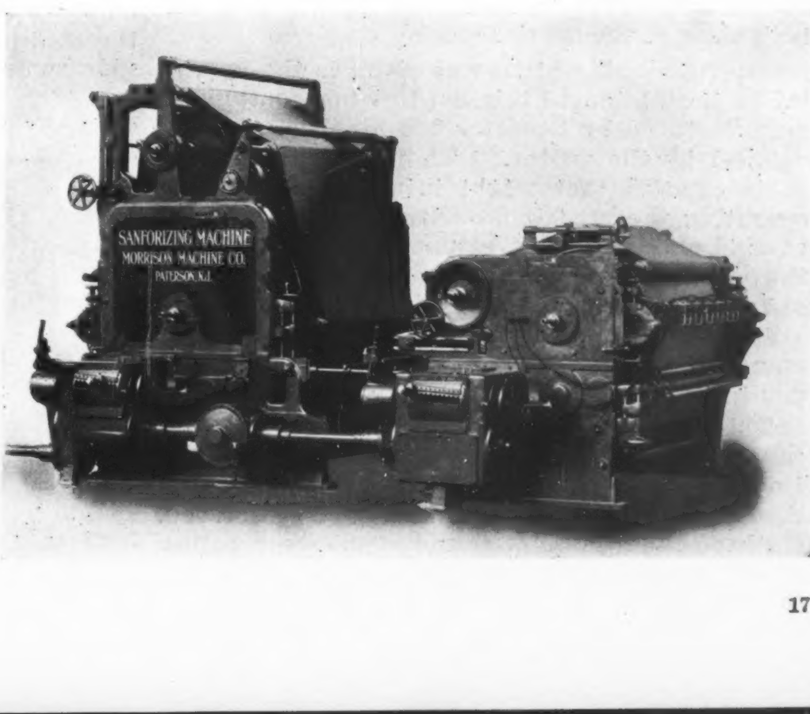


Fig. 1—Sanforizing machine considers the operator by enclosing all moving parts, simplifying operations and advancing automatically

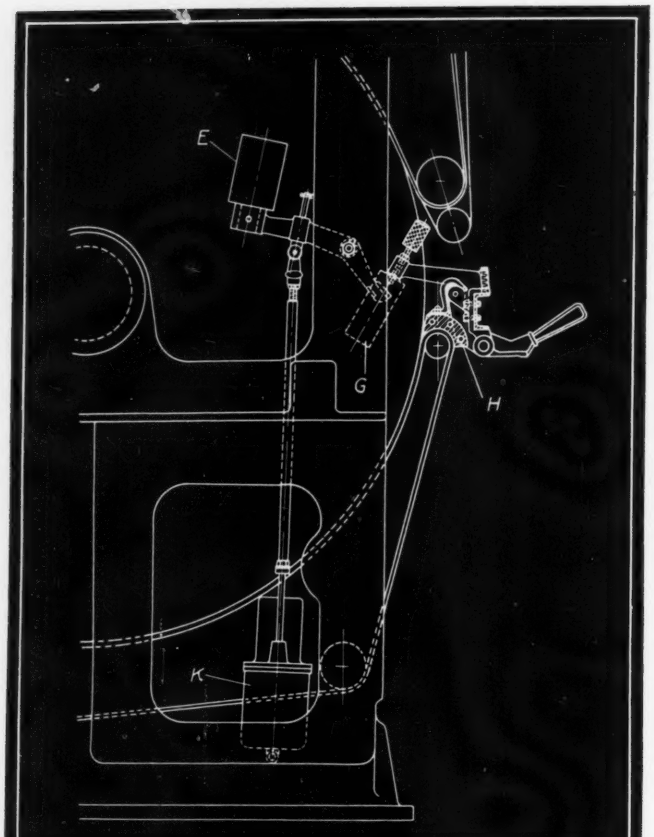


Fig. 2—Centrifugal force and gravity insure operation of safety device in the event of emergency

set forward of a steam and spray house through which the cloth passes to what is known as a modified Palmer machine, *Fig.* 1, left, supplied with a strategically arranged set of spreader and tension bars and electrically heated ironing shoes, with a large steam heated polished steel cylinder enveloped in an endless woolen blanket, with an auxiliary steam heated cylinder for drying the blanket. The cloth passes from the Palmer to what is known as a Duplex Palmer, *Fig.* 1, right, the cloth entering the Duplex Palmer reversed in order to secure an equal finish on both sides of the fabric.

Careful consideration was given to the operator in the design of this Sanforizing equipment manufactured by Morrison Machine Co. and developed by the writer. There are no moving parts exposed that might injure the workmen operating the machine. This precept has been carried out to such an extent that the shafts of the machine run inside metal tubing, thus not only providing a safety measure but also permitting their employment as steps should the operator wish to use them when making adjustments or threading through the cloth.

This enclosed design possesses further advantages in that it is possible to insure full splash lubrication for all of the major driving units. Appearance of the machine also is enhanced. Rather than a mass of frame members, rotating wheels and oil-smeared parts, the finished unit is a composite machine which expresses the cleanliness that it leaves with its product

The Palmer machine consists essentially of two large steam heated cylinders, one above the other, both of which are enveloped in an endless woolen blanket. On the top of this blanket there is located a series of shoes having a concave surface fitting the convex surface of the blanket over the feed roll in perfect contact. These shoes are equipped with heating units and are heated to a reasonably high temperature. As the cloth strikes these hot shoes in a damp condition, a very definite adhesion of the cloth to the blanket is secured. As the blanket is stretched mechanically in passing over the small diameter feed roll and the cloth adheres to it as it changes from a convex to concave position, which it does when it enters the reverse curve cylinder, the surface contracts where it before was expanded. The cloth contracting with this blanket causes a proportionate shrink in the fabric.

Because there is a loss in linear velocity of fabric between tenter and Palmer due to this shrinking, it is obvious that the Palmer must be made to run in relation to the feeder in direct proportion to the amount of shrink thus obtained. The necessity of providing a quick, positive method of adjusting the units was one of the first considerations taken up in the design.

Change Gear Box Adopted

It was decided to adopt a change gear box, *Fig.* 4, through which shrink steps of ¼-inch per yard could be obtained. This first redesign was a great step over the previous method which employed separate gears that had to be matched and installed each time a change in speed was desired, a particularly distasteful operation to the man running the machine as he had to be certain that the cloth was not soiled.

Exact relationship of parts was maintained by the frequent application of gearing and the wide employment of chain drives. In the use of the chain drives it was found necessary to provide some method of adjustment by which cor-

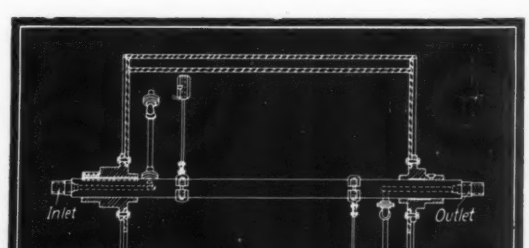
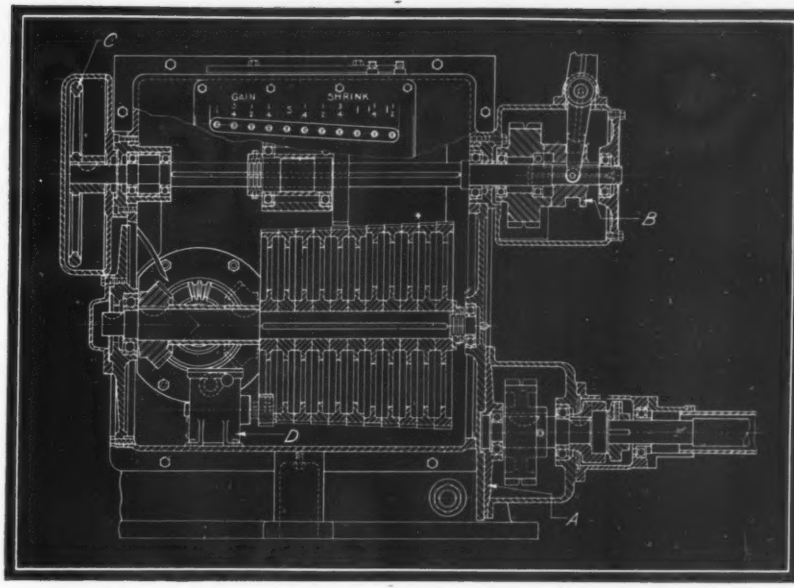


Fig. 3—Inner shell of steam cylinder reduces heat loss, aids cooling on the stopping of machine

Fig. 4—Eccentric plate permits the maintenance of proper tension on the connecting chain drive



rect tension could be maintained on the chain and yet insure rapid adjustment. This is accomplished by the eccentric plate in which is carried the ball bearing supporting the lower sprocket *A, Fig.* 4. Adjustment of this plate about the shaft permits the tension to be set to the right amount. Universal couplings connect the shafts so that no difficulty is encountered due to misalignment. These couplings also permit easy disassembly should this be necessary at any time.

The principle of no frictional engagements was also carried out in the design of the clutch employed to disconnect the Palmer unit from the duplex unit whenever it is required to back up one or the other for purposes of adjustment, threading, etc. The clutch *B* is a simple grab clutch, operation of which not only disconnects the machines mechanically, but also disconnects them electrically, thus performing all essential operations with one movement. The handwheel, *C, Fig.* 4, used in the backing up process can be seen at the lower right of the Palmer unit in *Fig.* 1. Its cover is closed during operation.

Manual Operations Eliminated

Elimination of another manual operation, and the accompanying possibility that the operator might neglect the step when it is most necessary, was accomplished by the development of an ingenious series of mechanism for use in the control of the heating elements. The elements themselves are thermostatically controlled, but should the machine be stopped for any reason, it is essential that they be removed from contact with the blanket covering the fabric. Should they remain for too long a time at one point due to the machine stopping, the blanket and the fabric are liable to be burned causing extensive loss if not a general fire.

The first step in the arrangement of control mechanism is the employment of a switch *D* directly geared to the shaft in the change gear box, *Fig.* 4. This switch operates through centrifugal force. When the unit is running at its proper speed, contact is made. This permits energization of thrustor, *K, Fig.* 2, and this thrustor raises counterweight *E* against the force of gravity. Whenever the machine is not operating, the weight of counterweight *E* holds a second counterweight *G* up, and with it the heating units, *H*, thus clearing them from contact with the blanket. When the force of counterweight *E* is removed, counterweight *G* brings the shoes into position and holds them with the correct pressure against the blanket.

Another problem was the development of a steam cylinder that could be used with high pressure steam and be perfectly safe, in spite of its large diameter, and one that would provide ample heat for the fabric, yet one that would not waste steam through radiation. One of the requirements for this cylinder is that when stopping work at the end of the day, the cylinder should be capable of cooling off as rapidly as possible so that production can be maintained up to the last possible moment. It is obvious that, if the blanket were left on a fully heated cylinder standing still, it would be badly scorched. Hence, the machine must be held in operation until the cylinder cools down.

This problem was solved by the development of what amounts to a cylinder within a cylinder, *Fig.* 3. The cylinder is made completely of steel, electrically welded. Its peculiar construction makes it safe, free from leaks, comfortable to work around, and easily heated with the least amount of steam. Between the double walls the steam is confined. This comparatively small steam space prevents the rapid expansion of steam, preserves its high temperature and reduces condensation to a minimum.

Fabrication of this inner shell by welding from the inner side necessitated openings in the end plates in order to get to the work. These openings, however were subsequently closed with sheet metal covers, converting the interior of the cylinder within the inner wall to dead air space, which, being an excellent nonconductor, largely prevents the radiation of heat from the ends of the cylinders and forces it out through the blanket where it is most effective. A heavy bronze bucket confined between the two cylinder walls insures rapid and adequate discharge of

the condensate. Two of the openings in the end of the cylinder are fitted with easily opened shutters. During operation of the machine they are tightly closed. When it is desired to cool off the cylinder at the end of a working period, the shutters are opened, fresh air rushes into the cylinder, the amount of radiated surface thereby being vastly increased and the rate of cooling speeded up considerably.

Strategic location of all operating devices and grouping of controls so that operators move in a minimum area were incorporated in the design of the Sanforizer. Grouped control on all units will be obvious from the illustrations, the controls being mounted on one side of the machine so that the side opposite may be conveniently placed in the closest relation to wall, columns or other machinery.

Automatic cut-offs are arranged to stop any overruns provided with power adjustments from injuring any of the mechanism, precluding any damage that might occur from forgetfulness of the operator.

Due to the inclusion of all operating mechanism on one side of the machine, it may be set in a minimum of space. It has been arranged so that it may be made in either right or left-hand design with equal facility. This means that two machines could be set in juxtaposition in a minimum of space, with the controls on both machines in convenient relation to each other.

The Sanforizers are equipped with a number of conveniences such as locking devices for all adjustments so that, adjustments having been made, they cannot possibly be dislodged from vibratory action of the machines. Lubricating facilities and reservoirs for copious supply of lubricant for bearing, gears, etc. are provided so that the machine requires only monthly lubricating attention.

New Books Discuss Design, Elasticity

Design of Machine Elements

By Virgil M. Faires; published by the Macmillan Co., New York; available through MACHINE DESIGN *for* $4.00 *plus* 15 *cents postage.*

Introducing the first chapter of his book with a quotation from MACHINE DESIGN, Prof. Faires launches upon a subject that intrigues every designer of machinery. In organizing his material he has maintained the conventional sequence, departing only where he has deemed it necessary in order to present at the outset those elements of machines in which the design requires the less complicated theories of the strength of materials.

Chapters then advance into design of elements which involves the more complicated considerations of the strength of materials. As Prof. Faires asserts, the most obvious advantage of this order of compilation is that the book is particularly well adapted to those courses in design that are begun by a student before strength of materials is studied, or which begin with and for a time run concurrently with the study of that subject.

This book is brought to the attention of the practicing engineer as a reference work from which he can review certain fundamentals that might be hazy in his memory. Chapters cover, besides materials and their properties, stress analysis, tolerances and allowances, screw fastenings, riveting and thin shell cylinders, belts, gears, chains, shaft design, bearings, springs, brakes, cams, welding, etc. The author is professor of mechanical engineering at the Agricultural and Mechanical college of Texas.

Theory of Elasticity

By S. Timoshenko; published by McGraw-Hill Book Co. Inc., New York; available through MACHINE DESIGN *for* $5.00 *plus* 15 *cents postage.*

With the more extensive application of the theory of elasticity in solution of engineering problems, there is a genuine need for a book such as this one prepared by Dr. Timoshenko. Keeping in mind practical applications, he has omitted matters of more theoretical interest and those which have not at present any direct use in engineering, in favor of the discussion of specific cases.

To simplify the presentation of data, the volume opens with a discussion of two-dimensional problems and only later when the reader has familiarized himself with various methods employed in solution to problems of the theory of elasticity does the author discuss three-dimensional problems. Mathematical derivations are put in elementary form and usually do not require more mathematical knowledge than is given in engineering schools.

In cases of more complicated problems all necessary explanations and intermediate calculations are given so that the reader can follow without difficulty through all the derivations.

SCANNING THE FIELD FOR IDEAS

TO QUOTE a prominent automobile manufacturer: "One quality of true ideas is that they cannot be hoarded. They must circulate; they must be shared. . . Nothing is more profitable for all levels of our life than nutritious ideas. Whoever gathers and circulates them, putting them easily and persuasively within reach of multitudes is serving the markets as well as the minds of men."

Circulation of ideas is precisely what MACHINE DESIGN aims to accomplish in its field. How this factor ties in with "long distance" engineering (to borrow a term from another automotive executive) is obvious. Without a generous store of ideas a long range view in new machinery development could not be attained. This latter executive says, for example, that his company designs engines far enough ahead so that they always have next year's unit ready to go. That means an even more intensive capitalization of ideas.

OBTAINING HIGH CLAMPING PRESSURE

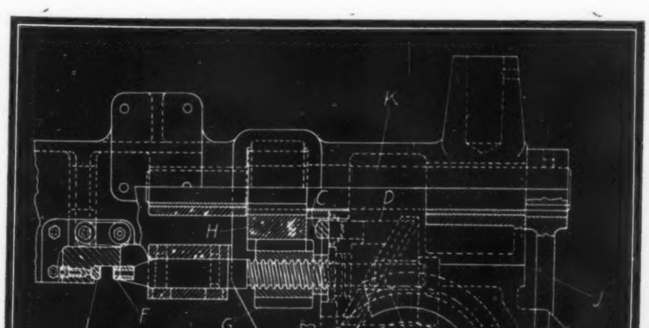
HERE is one solution to the problem of devising a mechanism that will provide rapid initial movement followed by a slower movement during which high pressure is generated. The arrangement is employed on the American Paper Bottle machine (M.D., April, p. 15). With this squeezing vise the rib at the top of the bottle is sealed under 865 pounds pressure.

In the operation of the mechanism, *Fig.* 2, cam *A* actuates rocker arm *B*, on the upper end of which is plunger *C* that forces block *D* downwardly. Block *D* slides on inclined stationary block *E* to effect rapid end motion to movable squeezing jaw *F*. During rapid traverse, pinion nut *G* slides in the teeth of gear segment *H*. When this operation is completed a remote cam with which *A* is synchronized, comes into action and through lever *J* rotates shaft *K* to turn gear segment *H* and thus rotate pinion nut *G* to obtain the desired additional pressure on the bottle rib interposed between the jaws *F* and *L*.

WINDING HELICAL SILICA SPRINGS

BECAUSE of their ability to measure small forces with great accuracy, helical springs of silica are finding more extensive application in scientific work. In the past these springs, drawn to the fineness of a thread, were made by hand, an operation that was difficult and tedious. Now, however, a better spring can be made by a machine that automatically winds the silica fiber into a helic, *Fig.* 1.

A recent issue of the *Bell Laboratories Record* discusses this process. The long straight



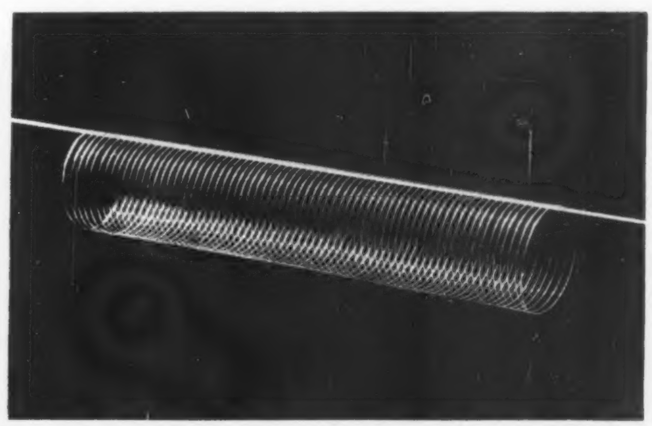


Fig. 1 *(Above)—Helical spring of silica is finding wide use in scientific work because of its ability to measure small forces*

Fig. 2 *(Left)—Mechanism provides rapid initial movement, followed by slower feed during which high pressure is applied*

fibers to be formed in spirals still are best drawn by hand. A clear silica rod is heated in an oxygen flame until soft. It then is removed from the flame and one end is drawn out quickly to the desired length. In threading the fiber into the machine, it is passed first through a tension device and then through a guiding slit. The mandrel to which it is subsequently fastened and on which it is formed, is a smooth silica tube turned by a motor through a thousand-to-one reduction drive.

The fiber is softened by flames from a double burner. One flame is adjusted to strike the fiber exactly where it comes into contact with the mandrel; the other flame is directed on the fiber and mandrel opposite the feeding side. Together they heat the mandrel uniformly over one-third of its circumference, thus preventing kinking of the fiber and giving it all the annealing it requires. Tension device, guiding slit and burner are mounted on a carriage which moves parallel to the axis of the mandrel while it turns. Pitch of the resulting spring is of course determined by the relative speed of the mandrel and carriage. Sensitivity of the springs, measured as their deflection per unit weight, increases rapidly with decrease in diameter of the fiber, and with increase in the diameter of the spiral and the number of turns.

GEAR TEETH DESIGNED TO FLEX

CAST iron consisting of 40 per cent steel, 1.0 chrome and 2.0 per cent nickel is used in what is termed a flexible tooth gear having epicyclodial addendums, with radial flanks. Depicted in *Fig.* 3, they are designed so that three teeth are in working contact at all times. Each individual tooth is said to flex a fraction of a thousandth of an inch at the time it comes in

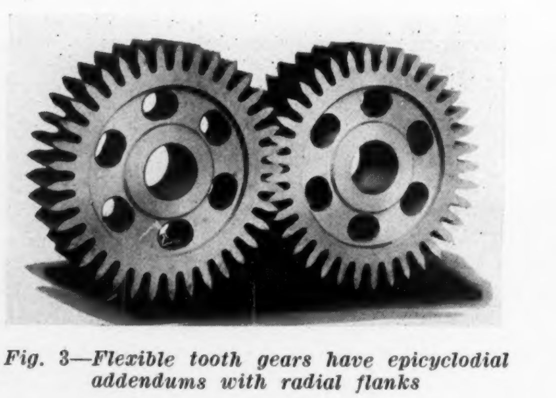


Fig. 3—Flexible tooth gears have epicyclodial addendums with radial flanks

contact with its mating tooth. Pressure is distributed among the three pairs of teeth which are theoretically under load. Thus a better oil film is maintained and shock and acceleration forces are eliminated. The gears shown are 36-inch pitch diameter, 16-inch face and weigh 5100 pounds. Roots-Connersville Corp. is using them in its larger size machines, both blowers and gas pumps.

COMPRESSED AIR CONVEYS MATERIAL

COMPRESSED air applications such as in a machine for applying concrete, refractory material, etc., reflect the increased use of pneumatic power in design. The unit, *Fig.* 4, handles all materials dry up to the point of applica-

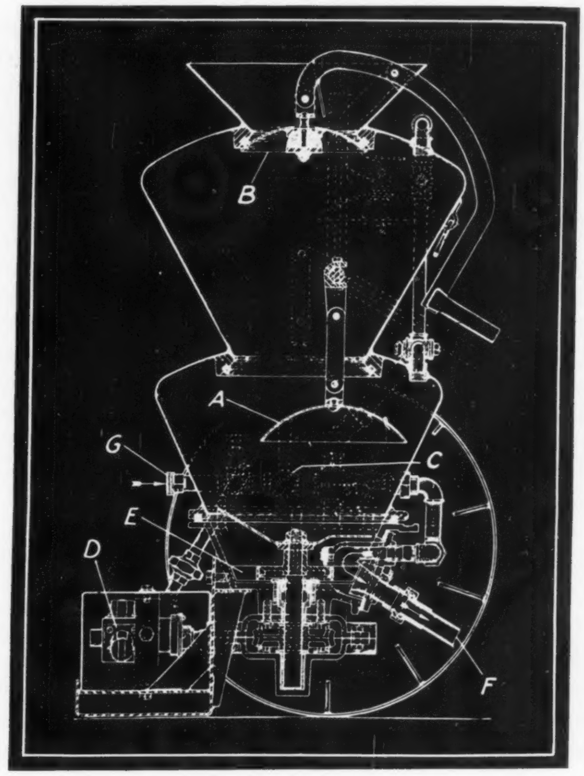


Fig. 4—Compressed air carries material in suspension through hose to point of application

tion, water being united with the mix at the nozzle. A constant flow of dry material passes through the hose in suspension and is blown along by air in this machine, the Gunitor.

After the lower bell valve *A* and upper valve *B* have been opened the machine is filled to the top with material. When this is completed and the valves are closed, to make the chamber airtight, supply valve *C* is opened. A motor *D*, either air operated as shown in the drawing or electrically operated, drives a feed wheel *E* which is like a gear with long thin teeth. Material placed in the machine rests on top of the feed wheel and particles drop into the pockets between the teeth.

At one point in the bottom of the unit under the edge of the feed wheel is an opening for connection of the material hose *F*. Coming through the side of chamber above this opening and over the feed wheel is a curved pipe connected to the air supply *G*. Air blows through this pipe, down

through the pockets of the feed wheel, and out through the opening to the material hose. As the feed wheel rotates each pocket filled with material passes through this air stream. The air picks up the material and carries it out through the hose.

EXPANDING RUBBER FOR INSULATION

A UNIQUE material is Onazote or expanded rubber insulation. It is light, an insulator against noise, resilient, high in dielectric strength, easily moldable to any desired shape and vermin proof. This vulcanized rubber product is expanded by gas under pressure. The material thus formed, when viewed under a microscope, shows a structure made up of minute cells bounded by rubber membranes. A paper summarizing information concerning this product was prepared by Harry D. Edwards, Union Carbide Co., New York, and published in a recent issue of *Refrigerating Engineering*.

MAGNET ACTUATES VALVE MECHANISM

FREQUENTLY an application of an electromagnet comes to light that embodies a clever idea. Such is the case of a new electro-hydraulic valve, *Fig.* 5, with an inlet designated as *A*. Electric current is employed to energize a magnet

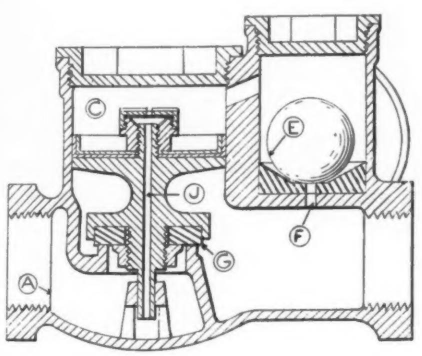


Fig. 5 — Magnetism is employed to control steel ball by which valve is opened and closed. Line pressure also plays a part in the operation

which throws ball *E* off bypass *F*, relieving pressure in chamber *C*. Bypass *J* being smaller than bypass *F* causes line pressure to raise seat *G* to allow the liquid to pass through the unit. When the current is turned off the ball rolls back on its seat, thus closing the valve. Myers Engineering Equipment Co., St. Louis, developed the mechanism.

FLOATING POWER PRINCIPLE EXTENDED

FEW industries exert as potent an influence on design in other industries as the automotive group. Here the interest of everyone is drawn to the new developments by the universal use of the product; the ability of the design staffs, and the widespread publicity and advertising of the manufacturers. Considering the research and engineering endeavor alone, it is little wonder that the ideas developed find wide application on other machines.

Now full floating power units have appeared in the oil burner industry. Not only have they appeared, but they also make use of coil springs,

Fig. 6—Three small coil springs are utilized to eliminate operating noise and vibration

those ever-reliable parts which first made the headlines with the introduction of independent springing. The Korth oil burner, *Fig.* 6, manufactured by Carter-Korth Oil Burner Corp., Roselle Park, N. J., employs three small coil springs to give 100 per cent floating power, thus eliminating operating noise and vibration.

The motor, fan, pump and pressure valve are suspended in air by these springs and even the electrode and nozzle assemblies float. In this manner an always troublesome vibration problem in burner design has been efficiently solved. Rubber washers are used at each end of the springs themselves to further eliminate noise and vibration. Cast aluminum aids in reducing weight, while a three point floor support assures ease of adjustment to the correct floor position regardless of surface irregularities.

NATURE YIELDS DESIGN IDEAS

LESSONS learned from close study of birds have in some respects been followed by Igor Sikorsky in airplane design. According to this noted engineer, water-going birds, particularly the large ones that fly over long distances in stormy weather and consequently have a very heavy wing loading. Commenting on the advantages of heavy wing loading to either bird or plane he declares that they feel bumps or rough air much less. When a gust comes along and hits them from below they bounce up only a little way as it were, whereas the lightly wing-loaded bird or plane is deflected upward considerably and has a long way to drop back again.

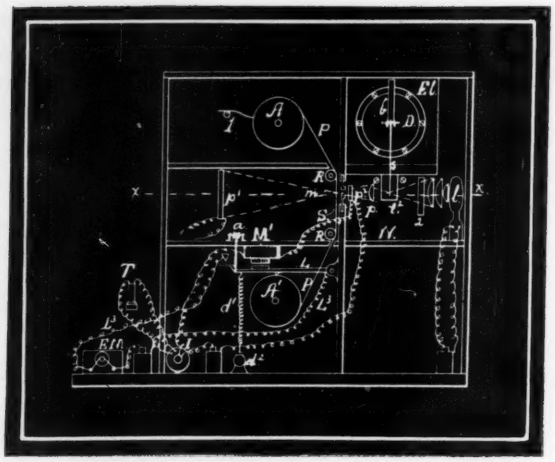


Fig. 1—Bargaining process for obtaining talking picture patent consumed thirty-six years

Patents Should Be Pr
Undue Delay

ON OCTOBER 22, 1880, Charles E. Fritts, an inventor, filed a patent application in the United States Patent Office for a method of recording and reproducing sound by a photographic process, *Fig.* 1. It was a meritorious invention. On October 31, 1916, after a period of more than thirty-six years, the patent office granted the Letters Patent. At that time, patent office records showed that 1,583,011 other applications had been filed during the pendency of this case. Even the inventor did not live to see his patent, the prosecution being carried on by the administratrix after his death. The records further showed that, besides this case, there were up to this time approximately 40 cases which had been in the patent office for fifteen years or more, and possibly 250 cases for ten years or longer.

The foregoing instances, of course, do not represent the ordinary condition, but rather the extraordinary. Probably the average time required to obtain a patent is from two to four years, but no patent attorney can ascertain this interval in advance with any degree of accuracy. A great deal depends upon the nature of the invention and upon whether or not the inventor through his attorney fights for all claims to which he is entitled in view of prior inventions. The prosecution of a patent application is in the nature of a *bargaining* proposition between the patent office, representing the general public, and the attorney, representing the inventor.

Not unlike any other bargaining proposition, the inventor will not receive any more patent protection than he seeks. The patent office is under no duty to aid the inventor, nor will it prosecute the case for him. The length of the bargaining period may be long or short. It may be less than a year if the inventor through his attorney yields prematurely to the demands of the patent office or fails to seek all the protection to which he is entitled. Such hasty practice usually results in a poor patent, and the unfortunate part about it is that there are a large number of inventors who never know whether or not they have a good patent. They think that, since their application went through the patent office in a hurry, they have a wonderful patent. This is not true. The best way to obtain a good patent is to proceed carefully and do a good job as one goes along.

The patent office confers no favors upon any particular inventor. It treats the rich and poor alike. Patent applications cannot be examined out of their regular order except in accordance with the provisions of certain special rules. Thus preference may be given to applications wherein the inventions are deemed of peculiar importance to some branch of the public service, particularly during national emergencies.

Typical Procedure Outlined

The following procedure briefly shows the actual steps that were taken in the prosecution of the electric appliance plug shown in *Fig.* 2. The invention of this plug resides in the fact that it releases the grip on the appliance terminal pins when the switch is turned "off," and thereby facilitates easy removal of the plug:

Dec. 30, 1930.—Inventor filed in patent office his application with 20 claims, covering both gripping feature and switching features. Initial filing fee

THE growing importance of patents in today's system of industrial competition, and the enlightening and entertaining manner in which Mr. Woodling has covered this phase of design work in the current and previous issues, have made each of these articles on patents one of the high spots of the issue in which it appears. A list of previously published articles on this subject appears on page 42.

Prepared with Care but Without

By George V. Woodling

$25.00. (This has since been increased to $30.00.)

Aug. 21, 1931—Patent office examined application and declared a division between gripping feature and switching feature, contending that there are two separate inventions, which, according to patent office rules, must be covered by separate patents.

Sept. 24, 1931.—Inventor replied to patent office (this is called an Amendment). The amendment pointed out why a division was improper on the ground that the gripping feature and switching feature function as a unitary structure, and therefore only one invention. (Amendments must be filed in six months from the date of last patent office action.)

March 16, 1932.—Patent office examined application in view of the amendment, withdrew the requirement for the division, and gave an action on the merits, rejecting all of the 20 claims in view of prior patents.

Aug. 9, 1932.—Inventor filed his second amendment, canceling 18 of the original 20 claims, and adding 9 new claims. This left 11 claims in the case. The amendment also included an argument supporting the allowance of the claims.

Dec. 16, 1932.—Patent office examined application of inventor's second amendment, and allowed 4 claims, rejecting 7.

Feb. 15, 1933.—Inventor filed his third amendment, adding 11 more new claims and presenting an argument supporting the allowance of all the claims, which now total 22.

May 1, 1933.—Patent office examined the application in view of inventor's third amendment, and allowed 13 claims, rejecting the other 9.

Oct. 4, 1933.—The inventor filed his fourth amendment, canceling the 9 rejected claims, and adding 2 more new claims. This left 15 claims in the case. The amendment also included an argument supporting the allowance of the claims.

Dec. 13, 1933.—Patent office allowed application with 15 claims.

June 8, 1934.—Inventor paid the final fee of $30.00.

July 10, 1934.—Patent office granted the patent.

Time required to obtain patent, approximately three and one-half years.

It can be seen from the foregoing illustration that, during the early stages of the prosecution, the patent office allowed no claims, but that by the bargaining process of writing new claims and presenting arguments in their favor, the patent office finally allowed 15 claims. Of these claims, some are broad, some are specific, and other vary in degree between the broad and specific ones, the broad ones being so broad that no one else beside the inventor can manufacture, sell, and/or use an appliance plug having the combination of a gripping and a switching feature without infringing one or more of the 15 claims. This means that the patent affords strong protection.

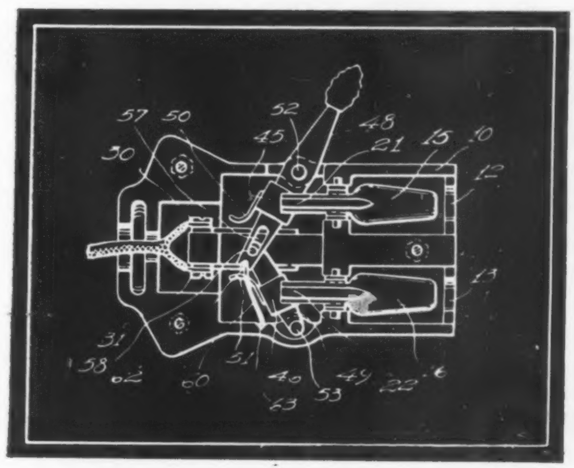


Fig. 2—Application for this switch was amended as often as new reasons for rejection were given

The inventor has the right to amend his ap-

plication as often as the patent office presents new reasons for rejections. In so amending, the inventor must clearly point out all the patentable novelty which he thinks the case presents in view of the prior patents cited by the patent office. However, the inventor cannot add new matter to the case. This must be covered by a separate application. An inventor can stop the bargaining process just as soon as the patent office allows one claim. This usually results in a weak patent because, should this one claim be subsequently declared invalid by a court, the entire patent may be ineffective. On the contrary, the inventor may unduly prolong the bargaining period. This is what Fritts did. Manifestly, this is against the policy of the patent office.

It must not be inferred, however, that the patent office is free from fault. A certain portion of the delay is directly chargeable to it. Note

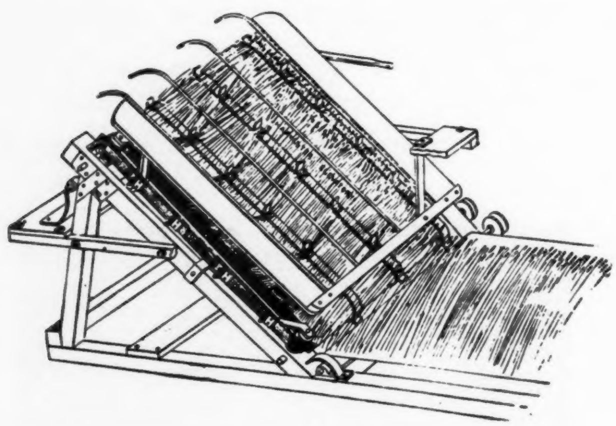


Fig. 3—Narrow patent claim opened field for competition in this harvester improvement

that it was approximately eight months from the filing date of the appliance plug application before the patent office made its first reply. This is mostly due to the fact that there is a lack of sufficient help to handle the applications as fast as they come in, thus working a hardship upon inventors.

But on the other hand, strange as it may seem, there are some inventors who purposely seek delay. Why should they? There is no protection until the patent is granted. Thus, should a competitor begin to manufacture and sell the inventor's device there is no recourse, so long as the inventor's patent application is still pending. The competitor may even flood the market without being enjoined or sued for damages. Neither are the rights created by the issuance of his patent retroactive, so as to reach backwards and catch his competitor who manufactured and sold the invention during the pendency of his application. Ostensibly, an inventor should avoid delay, but actually there are many advantages to be gained by delay which greatly outweigh the disadvantages of not being protected while the application is pending. The most obvious reason for a delay is to make the seventeen year term of the patent life coincide with the manufacturing period which produces the maximum financial return. This is particularly true where the invention is born before its time.

Another reason for delay is to give the inventor sufficient time for experiments before his patent issues. This enables the inventor to work hand-in-hand with his attorney to secure a strong patent. As a result of experimental development it may be found that some of the main features of the original invention may be dispensed with, at the same time giving better operation at reduced cost. If such turns out to be the case, these facts should be related to the inventor's patent attorney, who then may draft new claims to cover the invention broadly, leaving out the dispensable features. This makes a strong patent. To design around a strong patent it is necessary for competitors to eliminate at least one part over and above what the inventor had already eliminated by his own experiments. This may be difficult for the competitor to do, if the inventor had already reduced his invention to its lowest terms.

Protect Separable Parts Separately

There is another advantage growing out of such experiments, in that the inventor may discover that one or more of the parts of the entire machine may be useful in a different environment from the original machine. Here again the inventor should bring such facts to the attention of his patent attorney. The proper practice, in cases of this nature, is to protect the separate part by a separate claim or claims. This enables the inventor to enjoin a competitor from using this separate part. The invention of a blade is one of the essential elements of a safety razor. But there are many possible mechanisms for holding the blade. Therefore, to obtain the maximum protection, the blade should be claimed alone, and not in combination with the holding mechanism.

Besides affording an opportunity for experiments, delay makes it possible for the inventor to analyze the construction of competing devices and to make his patent cover them. He cannot, under the rulings of the patent office (except for special reasons) obtain access to other patents pending, but often can unearth valuable information by keeping close watch on activities in his field. The following example will illustrate the importance of this vigilance.

On Nov. 17, 1877, an inventor by the name of William F. Olin filed a patent application for an improvement in harvesting machines, *Fig.* 3. On Jan. 27, 1880, a little over two years later, his patent was duly granted with very narrow claims. In harvesting machinery at that time,

and even up to the present time, the grain as it is cut falls upon a platform, and is carried to the base of an endless belt or canvas provided with teeth or slats which seize the grain and carry it over the driving wheel of the harvester, up to a higher level than that where the binding is done, from which point the grain falls a short distance to reach the binder mechanism. This worked fairly well, but by reason of the fact that the butt ends of the stalks were heavier than the heads, there was a tendency for the heads of the stalks to be delivered to the binder mechanism in advance of the butt ends, thus interfering with the proper binding of the grain. To obviate this, Olin invented an auxiliary belt located with the axis of its pulleys at a right angle to those of the main belt, but with the auxiliary belt moving in the same direction as the main one, and at a somewhat higher speed. This kept the butt ends up to a level with the heads.

Failure to Claim Is Dangerous

After Olin obtained his patent the defendants manufactured and sold a harvester almost identical in principle with the harvester described in the Olin patent, except that they located the auxiliary endless belt on the descending side, *Fig.* 4, which adjusted the grain in an even fashion just prior to the grain's entering the binder mechanism. Upon the strength of his patent Olin sued the defendants for infringement. The real question raised in this case was whether the narrow claims of the Olin patent describing an auxiliary belt located on the *ascending side* can be construed to cover a similar device located upon the *descending side*. The court was of the opinion that it did not. The court pointed out that, while an inventor may be entitled to a broader patent than he obtains, he is presumed to have abandoned the residue to the public if he fails to claim all the features to which he was originally entitled. This case may appear to be unfair to Olin, but it was his own fault that he obtained a weak patent.

There is something in common to both the plaintiff's and the defendant's devices, in that they both employed *an endless belt moving in the direction of the grain delivery, operating in contact with the butt ends of the stalks to deliver the grain to a binding mechanism.* This is the broad principle of Olin's invention and had he not prematurely yielded in his bargaining and thus kept his patent application pending a little while longer, he would have had a chance to present claims embodying this broad principle, and upon their allowance by the patent office he could have successfully won his case against the defendants. Thus it pays to bargain unyieldingly until the inventor obtains all the protection that he is entitled to, even though it may take several years longer to obtain the patent.

It is not advisable, on the contrary, to extend needlessly the bargaining period after the inventor has obtained his full measure of protection. This constitutes an evil and is repugnant to the purposes of the patent laws, in that it delays, instead of promoting, the progress of science and inventions. In the Fritts case there were approximately two hundred claims presented to the patent office. Claim 187, as well as several other late claims, was admittedly introduced to cover the entire art of producing modern moving pictures. The patent office, however, refused to allow these claims, pointing out that these long cases, in effect, constitute hidden traps for later inventors who, without knowledge of their existence, in good faith produce similar inventions and apply for patents only to find their efforts anticipated and their labor lost.

For individual inventors who wish to sell their inventions there is another reason for seeking delay, and that is that large concerns prefer to buy patent applications instead of patents. Their reason for this is that the attorneys for the large concerns can take over the prosecution of the case and make a strong patent out of the application which might otherwise have matured into a weak patent in the event that the individual inventor had a poor patent attorney.

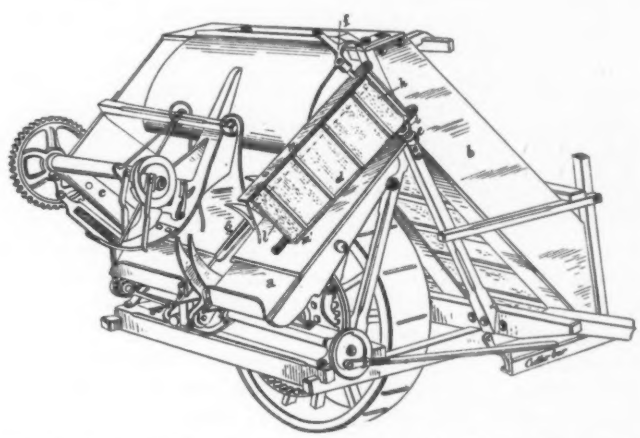


Fig. 4—Ingenuity in placing belt on the descending side circumvented claims of an earlier patent

In those cases where the invention resides in a field which is highly competitive and where the inventor wishes to manufacture and sell his inventions himself, it may be advisable not to manufacture and sell his invention until he first obtains his patents. This means that he would be in a position to enjoin his competitors upon the strength of his patent just as soon as they entered the field. It is important that, just as soon as the patent issues, the number be stamped or otherwise placed upon the patented device. The law provides that "It shall be the duty of all patentees, and their assigns and legal representatives, and of all persons making or vending any patented article for or under them, to give sufficient notice to the public that the same is

(*Concluded on Page* 42)

Bearing Metals, Lubrication Still Face Engineers

EVEN a panoramic view of S.A.E. activities at the recent convention reveals many forward engineering steps. Attention is drawn in the following, for example, to the technical discussion on engine types adapted to automobile design trends. Also to the discussions on lubrication, bearings, and trucks. They indicate what may be anticipated in automotive and possibly more generalized design.

EXPERIENCE has made the automotive engineer a pessimist. He designs his lubricating system to maintain a film of oil between the journal and bearing but realizes that he will not succeed and takes care to select a bearing material that will minimize the harmful effects of his failure. Babbitt is the material which for many years has seemed best suited to this purpose. With this statement Stanwood W. Sparrow, Studebaker Corp., opened his discussion on main and connecting rod bearings.

Babbitts differ widely as to analysis and physical properties. In one engine the best results were obtained with a babbitt consisting of 2½ per cent copper, 8 per cent antimony and 89½ per cent tin. In another engine satisfactory results were obtained with 3.7 copper, 7.3 antimony and 89 per cent tin. Results very nearly the same were obtained with 4½ copper, 4½ antimony and 91 per cent tin.

Why Babbitt Is Used

Despite a considerable amount of test data it would require some courage, Mr. Sparrow asserts, to state positively that any one of these three analyses is definitely superior to the others. The widespread use of babbitt may be due to the fact that metal-to-metal contact over a small portion of the surface causes the bearing material to crush without damaging the journal or communicating sufficient heat to the remainder of the bearing to destroy the oil film.

Modern production methods make it possible to use a thickness of babbitt of 0.015-inch or less. Mr. Sparrow declared that in his experience a thickness of less than 0.030 has been found undesirable. The thickness does not seem to have a major influence upon wear or cracking. With very thin babbitt, however, if a small piece breaks loose it can tip sufficiently to bring its edges in contact with the shaft. These are ground away rapidly and eventually the entire section disappears. With thicker babbitt, there is not likely to be any loss in material until the cracks extend to the edge of the bearing.

To some extent copper-lead mixtures are superseding babbitt for main and connecting rod bearings. It is generally understood that this material has a melting point considerably above that of babbitt and that its physical characteristics are less affected by temperature. The mate-

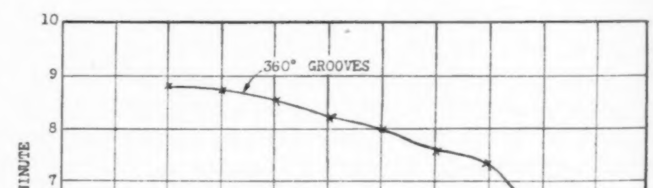


Fig. 1—Curves show oil flow in relation to length of main bearing grooves. Pressure 40 pounds, oil temperature 150 degrees Fahr., viscosity 108. The 360-degree groove is recognized as detrimental to main bearing performance but with it more adequate supply of oil can be delivered to connecting rod bearings

rial appears to crush as readily as babbitt and metal-to-metal contact produces a film of lead which presumably has sufficient lubricating value to prevent seizing. A typical analysis shows 45 per cent lead, 2 per cent nickel, with the remainder copper.

With reference to lubrication, Mr. Sparrow has prepared a chart, *Fig.* 1, which shows the effect of changing the length of the grooves in main bearings. At 4500 revolutions per minute practically no oil enters the crankpin when the groove in the main bearing is replaced by a single hole. It is not his intention to give the idea that the 360-degree groove is ideal for a main bearing. The purpose of the groove in almost every case is to supply oil to the crankpin and its presence is generally recognized as detrimental to the main bearing itself. The matter may be summarized, Mr. Sparrow sets forth, by saying that in general the grooving of main bearings is essential to the adequate lubrication of connecting rod bearings and that adequate main bearing life can be obtained in spite of grooves.

Trends in Automobile Engines

Outlining engine types adapted to automobile design trends, L. P. Kalb, Continental Motors Corp., brought attention to the radial engine. Although never applied to the automobile, the use of this type of engine in airplanes has become so general that other types can be considered as rare exceptions. The radial has two outstanding advantages, both of which render it desirable for aeronautical applications. The first and most important is its low weight, which of course would be a desirable feature for automotive use. It is estimated, according to Mr. Kalb, that in a radial engine for use in an automobile three to three and one-half pounds per horsepower could be attained without departing from conventional automobile engine materials.

The second inherent advantage of the radial engine from an aeronautical standpoint, is the fact that this cylinder arrangement lends itself to air cooling. This may not at present be considered an asset for automobile application, but at least with this cylinder arrangement some of the undesirable features of air cooling an automobile engine would be diminished.

Rear-Mounted Power Plants

Probably the greatest advantage of the radial engine for automotive use would lie in its compactness and structural arrangement, both of which adapt it particularly well for a rear engine installation. Compactness and low weight both tend to eliminate one of the particular disadvantages of the rear power plant, which is the overhanging weight behind the rear axle, *Fig.* 2.

Comments of Austin M. Wolf, prominent New York consulting engineer, on lightness in truck design were particularly timely. Deep frames required at the present time could be eliminated and the combined frame and body structure would total less than in the case of present construction with the unpardonable discontinuity of structure behind the cab except for two frame rails disposed at a most disadvantageous position. We all complained, he says, when the original vertical hydraulic hoist was instrumental in frame breakages at this point. Yet we continue today with this same "barbaric" frame construction. Aluminum alloys could be used to advantage in a combined frame-body structure.

Height and width of the body-frame structure would provide panels that lend themselves to the

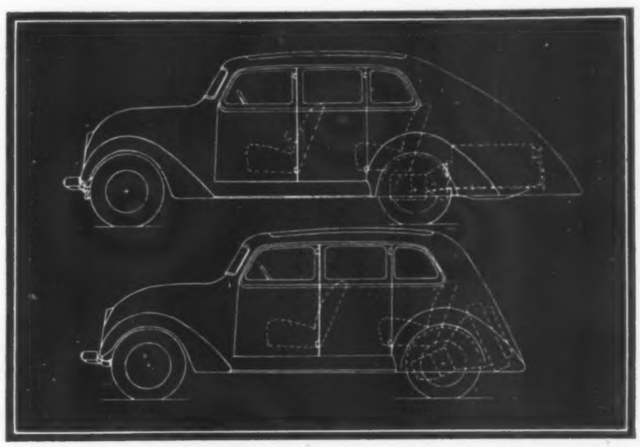


Fig. 2—Use of a radial engine for rear mounting in auto would reduce amount of overhang considerably as shown in bottom sketch

most admirable type of truss construction and afford high moments of inertia. The body rail forms the cab front and top outlines and continues into the body section proper. At the front the sloping windshield pillar or body rail joins the chassis portion of the frame ensemble. Mr. Wolf predicted that in the future as speed is increased, we can expect to see an end to open-top truck bodies in order that the load be kept within the proper minimum wind-resistance form.

Much interest is being displayed in glass fiber, popularly referred to as glass wool or glass cotton. According to Arthur D. Little's *Industrial Bulletin* the finer grades are employed at present largely for heat insulation. Glass wool is completely fire and vermin-proof, while the resiliency and length of the glass fibers practically eliminate the tendency to settle and leave open spots at the top of the insulated space which so frequently occurs with short-fibered mineral wools, especially where vibration is present. Accoustically the insulating value of this material is excellent.

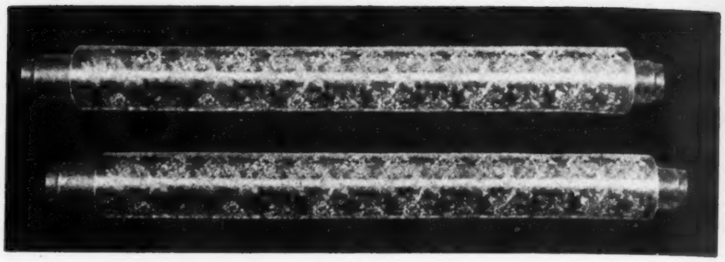


Fig. 1—Resistance of plating to wear preserves expensive engraving on cloth printing rolls

Hard Facing, Inserts and Plating Satisfy Hardness Requirements

By Allen F. Clark

CHROMIUM plating, long an able ally of the designer desiring lustrous appearance, is daily finding new applications in an entirely different field, a field where its appearance is of importance only as a gage as to the efficiency of the plating process. These new applications are concerned with wear resistance and the products plated embrace integral parts of machines as contrasted with trim, cases and accessories, the group formerly constituting the major portion of the applications. An example of the relative unimportance of appearance in this new group can be found in the application of chromium plating to piston pins, a part as thoroughly hidden in the finished machine as any that can be imagined, yet one where the plating has been found to be of exceptional value to the design. Here the sole purpose of the application is to resist wear, a design problem also discussed in the articles on hard-facing rods and tungsten carbide inserts in the May and June issues of MACHINE DESIGN.

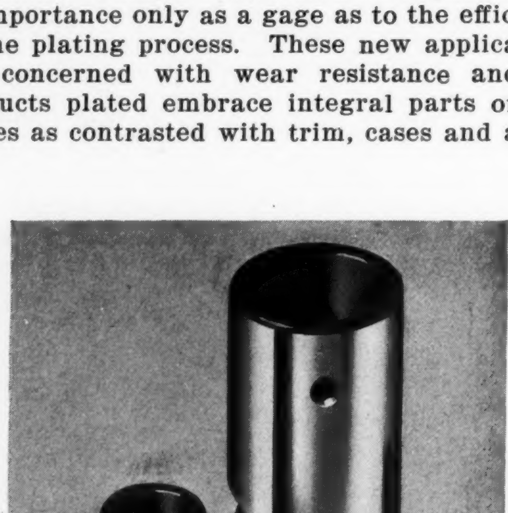


Fig. 2—Corrosive and abrasive action on piston pins is resisted by thin coating

The wide-spread application of chromium plating and the dissemination of information on the process has, in a general way, made the production steps and appearance value well known. However, in applying the finish for machine parts, especially those required to withstand wear or abrasion, it is well to understand the properties of the plating and the uses to which it can be put.

Resists Action of Acids

Chromium, the pure element, has a hardness of 9 on Moh's scale of hardness. On the same scale the diamond is 10; boron is 9.5; corundum is 9; manganese is 5 and iron is 4.5. Electrodeposited chromium is harder than any form of case hardened steel, and it is extremely tenacious and adhering when properly deposited on steel. The coating, when properly applied, resists the action of organic sulphur compounds, sulphur dioxide, or hydrogen sulphide even at elevated temperatures. It is highly resistant to atmospheric corrosion, heat, wear and abrasion, including the abrasion produced by granular sub-

stances. It is unsoluble in nitric acid and alkalis as well as most organic acids.

Such properties as those outlined in the foregoing make the finish ideal for applications where both corrosion and wear resistance must be obtained. In addition to its use on gages, tools and dies, the plating has been applied to bearings, cams, valve seats, thread guides, gears, plate glass and paper rolls and similar machine parts. As the plating is unaffected by practically all food products, it can be freely used on parts which contact foods.

Base Metal Provides Strength

One important point in connection with applications that the designer should remember is that chromium plating should not be used as a substitute for high grade steels despite the hardness and wear resistance of the finish. The plating should be done over a hard surface as the mechanical strength of the part is entirely provided by the base metal. Plating can be done over welds, but care should be taken that the welds are properly smoothed over by grinding and are free from pinholes. These precautions are necessary to insure complete coverage since the plating will not adhere over oxides and will not cover defects in the base metal.

Care, of course, must be taken in the design of parts for chromium plating to insure ease of plating. Although the modern plater can overcome a great many difficulties, assurance of a satisfactory job can be obtained by the elimination of unprotected sharp points, deep recessess, re-entrant corners and similar oddities. Then too, the part should be designed to simplify racking, provision should be made for the escape of gases generated in the process and for ease of rinsing, and it should be remembered that parts designed so that the current density can be evenly distributed on all surfaces will insure uniform results at an economical cost. Of possible assistance to the designer planning the inclusion of simple parts in his machine is the availability of a commercial sheet which can be purchased already chromium plated. Such sheet can be formed by any of the ordinary commercial methods without damage to the plate provided the draws are not too deep.

Pins Lapped Before Plating

A detail in specifying parts for plating which should be of value to other designers is in the production of piston pins, one of the automotive parts manufactured by Thompson Products Co. In producing these pins, *Fig.* 2, plated to resist wear and corrosion, all of them are lapped after being finish ground. While the ground surface is one of the better machine finishes, it still leaves microscopic high points which, when plated, reproduce the action of a file on the pistons and thus quickly enlarge the opening, thus loosening the action and creating noise and inefficiency. The plate used on these pins is 0.0002 to 0.0003 inch thick, and its lustrous appearance is employed as a guide to its hardness. An incidental advantage of the finish is found in the fact that with the plated pins it is not necessary to pack them in waxed paper and grease when shipping. The plating so protects them that they may be merely packed in a box without any additional protection.

An interesting application of chromium plating on machine parts, although one not directly connected with wear, is its use on steel shafts running in zinc die cast bearings. Steel shafts frequently freeze to the bearing. Chromium plating the shafts completely eliminated this trouble.

Chromium plated water pump shafts, *Fig.* 3, are frequently employed on intermittently operated water pumps to prevent corrosion during idle periods, with consequent scoring of the bearings or complete freezing of the shaft to bearing. Plating the shafts has eliminated the need for replacement in some uses, except for occasional breakage.

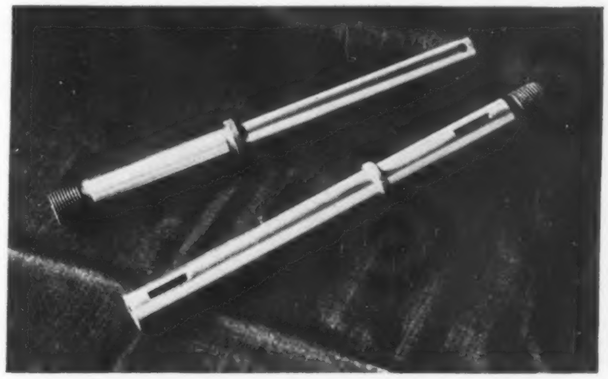


Fig. 3—Finish prevents corrosion of water pump shafts which operate intermittently

Shut Down Frequency Reduced

The textile industry has found wide use for the finish in resisting the wearing action of threads. Flyers and spinners are often subjected to this severe wear, and their redesign for chromium plating has considerably increased their life. Another textile machine part where the finish has proved of value is washboard tensions. Failure or necessary replacement of one of these parts would require the shutting down of the entire loom and result in a considerable loss of production time. Their finishing to combat wear has reduced the frequency of such shut-downs.

Liners of pumps handling ceramic sludges, blowers passing dusty or gritty substances and the walls of the cylinders on outboard motors are all machine parts where the application of chromium plating aids the fight against wear

and corrosion and increases the life of the machine. A somewhat similar application is the use of the finish on steam turbine blades where extreme erosion is combated.

The petroleum industry has found the combined resistance of the finish to oxidation by flue gases and to the sulphur compounds of petroleum of value in protecting stills and piping. One of the most serious problems in oil cracking equipment has been the corrosion of expensive reaction chambers. Various expedients have been tried but with little success. Special liners have been used but were found impractical due to seepage between the liner and the wall of the vessel. The resulting corrosion cannot be detected and is therefore very dangerous. Chromium to date has proved itself to be an exceedingly satisfactory remedy for the corrosion problem in this industry.

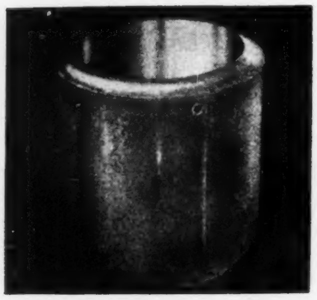


Fig. 4—Tank used in paper industry employs chromium to combat the action of alkalies

Protects Screen Plate

Resistance of the finish to the action of sulphate, sulfite and soda liquors as well as to wear and abrasion is the basis of its application in the paper industry. Few applications have aroused more interest than the use of chromium plate as a means of life insurance for screen plates, both flat and curved. The extremely smooth surfaces of plated screen plates (including the inside of the slots which is completely covered) prevent adherence of dirt and slime. Slots do not clog, little if any cleaning is required and there is freedom from metallic contamination of stock. A noticeable increase in quality of stock results from these improvements. Of equal importance is the maintained precision and permanent accuracy of the slots. Press plates used in the forming of fibre board are another group of paper industry parts where the action of corrosion and abrasion is resisted by this finish. A copper tank used in the paper industry, *Fig.* 4, was chromium plated on the inside to resist the attack of borax, ammonia, tri-sodium phosphate and other alkalies.

Rolls such as drying rolls, metal finishing rolls, calendar rolls, Schreiner rolls and cloth printing rolls amply repay the designer specifying a plating for them. They not only give rust protection, easier cleaning and longer life, but plated Schreiner rolls impart a permanently higher lustre to the cloth. Costly operating difficulties due to scratches and roll failures are avoided by the use of plated copper printing rolls for printing cloth, *Fig.* 1. The plate does not dull or damage the finest engraving. The printing surface is protected against corrosion; the designs are made sharper and clearer; lustre of the material is increased; and gums and foreign material are prevented from adhering to the plate. The raised portions are also protected from the action of doctor blade and ink.

Such applications as those outlined in the foregoing are merely a few of the many that have already been placed in successful service. They may suggest further uses to the designer facing the problem of combating wear, corrosion and abrasion.

Articles on this and allied subjects published in previous issues of Machine Design include:

"Overcoming Sales Resistance by Color Effects and Finishes," by William J. Miskella, March, 1930, p. 15.

"New Use of Chromium Plating," April, 1930, p. 21.

"Evolution in Finishing Processes Worthy of Designer's Study," by M. J. Callahan, May, 1930, p. 46.

"Effecting Economies in Finishes," by William J. Miskella, Nov., 1930, p. 41.

"Specify Proper Finish and Color for Your Product," by T. J. Maloney, Jan., 1931, p. 33.

"Specifying Close Limits Is Futile Without Suitable Finish," by R. E. W. Harrison, Feb., 1931, p. 46.

"Considering Machine Finishes from the Sales Standpoint," by William J. Miskella, May, 1931, p. 41.

"Is It Possible to Standardize Machine Finishes?" Dec., 1931, p. 22.

"Improving Design with New Finishes," by William J. Miskella, Oct., 1932, p. 37.

"Porcelain Affords Protective Coating," Nov., 1932, p. 23.

"Sprayed Metal Offers Possibilities," Dec., 1932, p. 21.

"Supplementing Mechanical Efficiency with Good Appearance," by Walter Dorwin Teague, Dec., 1932, p. 22.

"Metallic Coatings Combat Corrosion and Provide Attractive Finish," by Gustaf Soderberg, April, 1933, p. 18.

"Designer Should Specify Type of Machine Finish," by R. E. W. Harrison, Sept., 1933, p. 23.

"Consider Finish During Design to Reduce Costs," by S. P. Wilson, Oct., 1933, p. 31.

"Bright Colors on Machines Aid Operating Efficiency," by T. J. Maloney, March, 1934, p. 22.

Motor Drives Are Discussed

THE application, operation and maintenance of several types of motor drives are presented in a 224-page book entitled "Fifty Reports on Mechanical Power Transmission from Motor Drive to Industry." The book goes thoroughly into the drive problem, giving the results of experiments and applications. It includes studies of driven loads and how to belt them; studies of motors and motor loads and how to belt them; fundamentals of belt transmission; a comparison of short center drives; trouble jobs and how to solve them; modern group drives versus individual drives; and similar information. Copies of the book may be obtained from the American Leather Belting association, 41 Park Row, New York, for twenty-five cents.

Determining Capacity of Helical and Herringbone Gearing

By W. P. Schmitter

STEEL is used almost exclusively for helical and herringbone gears, discussed in the first section of this article beginning on page 40 of the June issue of MACHINE DESIGN. When the heat treating operations are properly carried out and good quality gear steels are used, brinell number may be used as a criterion of the capacity of the material. The relations existing between ultimate strength and brinell hardness are universally recognized. There is also sufficient conformity between steels of the type being considered to warrant using brinell hardness as an index to the yield point. *Fig.* 24 shows this relation. The yield point in shear is taken as two-thirds of the tensile yield point.

The endurance limit of steels varies to a greater extent, but up to a limit of 350 brinell it is safe to use the hardness number as an index of fatigue characteristics if we again limit ourselves to accepted gear steels. The curve in *Fig.* 26 is based to a large extent on experience, but it does agree fairly well with the values obtained

Fig. 24—Relations between stress and Brinell and Shore hardnesses

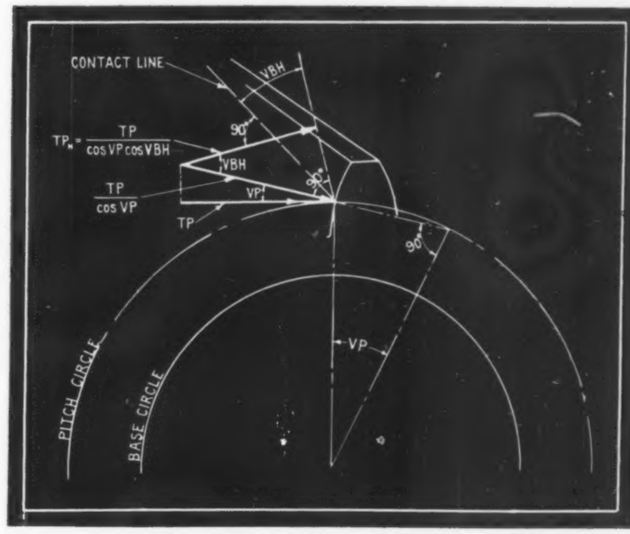


Fig. 25—The normal pressure on the contact line is a function of the pressure angle and the base helix angle

in reversed torsion tests at the University of Illinois. It would seem that a crystal in the compressed area below the surface of contact is first acted upon from one side, and then from the other in such a manner that the stress range is more severe than direct loading, but probably not as severe as a completely reversed stress.

The best way to determine the endurance limit for any gear steel is in a gear testing machine. Then if the gear formula shows the correct mode of variation of capacity, the correct rating will be obtained for any condition of ratio, velocity, size, etc. It is obviously impractical to do this for all gear steels; therefore, the general endurance curve must be used to show the trend.

For relatively soft gears, both the yield point and endurance limit must be considered. A gear must be designed to take a certain momentary overload, usually assumed at 100 per cent. This overload may occur during starting because of the additional accelerative forces, or it may occur during operation from one cause or another. If the load at this maximum designed overload capacity be in excess of the yield point then plastic deformation or some other form of surface failure is apt to take place. The yield point of the gear material must therefore be

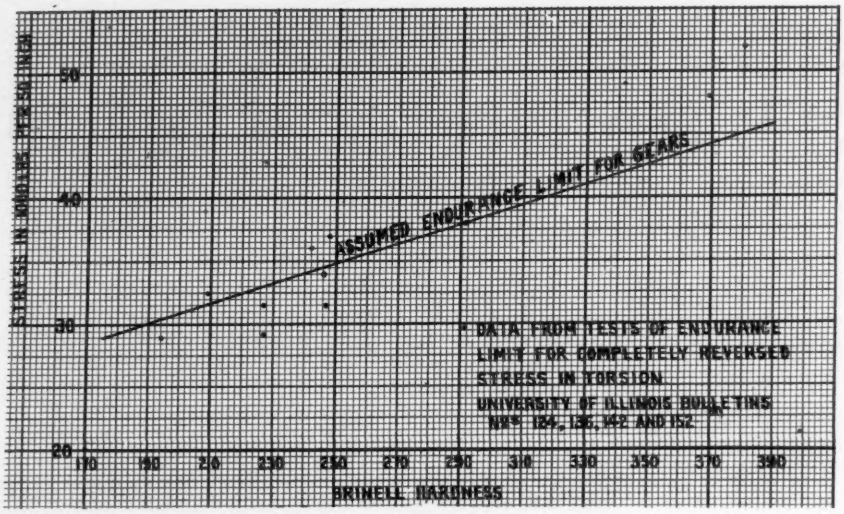


Fig. 26—Hardness number may be used as index of fatigue characteristics

considered in determining the allowable loading. For materials 200 brinell or less, the yield point in shear sometimes limits the rating because the endurance limit at the running load will not be reached in those cases.

For the harder materials, endurance limit is the prime consideration and it will be found that considerably more than 100 per cent overload can be momentarily applied without damage to the gear set, providing of course that the overload is not recurrent.

While the gear must be designed for both the maximum allowable overload and the normal running load, actual computation for the individual loads is eliminated. *Fig.* 27 shows the allowable shear stress for the gear based on the shear yield point and the endurance limit, whichever happens to be critical at the hardness considered. The value for the pinions is based on an allowable stress of $S_y \div \sqrt{2}\text{-}1/3$ at the lower hardnesses with an increase similar to that of the gear curve at the higher hardnesses. The pinion should always be harder than the gear in order to allow for work hardening of the gear, to avoid any possibility of seizure, to aid in the preservation of the involute tooth form, to allow for greater abrasive wear on the pinion, and to provide a greater measure of protection on the pinion for prolonged overload. This does not apply to materials of extremely high hardness.

It becomes increasingly difficult to obtain the optimum from materials as hardness increases. The hard gears do not have the same degree of self-corrective properties, and the finish is apt to be inferior. The endurance limit of the material diverges from the yield point as hardness increases. Also other physical properties, such as reduction in area, elongation and grain size, play an important part in determining toughness and impact values. Deflection becomes more objectionable at the higher hardnesses. All this has been considered in setting values for allowable working stress, but the necessity for tightening up on mechanical errors at these higher hardnesses should not be overlooked.

In the equation for elastic cylinders, the maximum shear stress was found to be

$$S_s = 958\sqrt{P' \times (r_1 + r_2)/2r_1r_2}$$

For helical gears we can sub-

EVEN though the formulas regularly used in the rating of helical and herringbone gears have proved successful, the confusion between the various systems has lent unusual interest to this article by Mr. Schmitter of the Falk Corp. The first section appeared on page 40 of the June MACHINE DESIGN. Comments by authorities in the gear industry appear on page 61 of that issue and page 52 of this, while nomenclature was listed on page 59 in June.

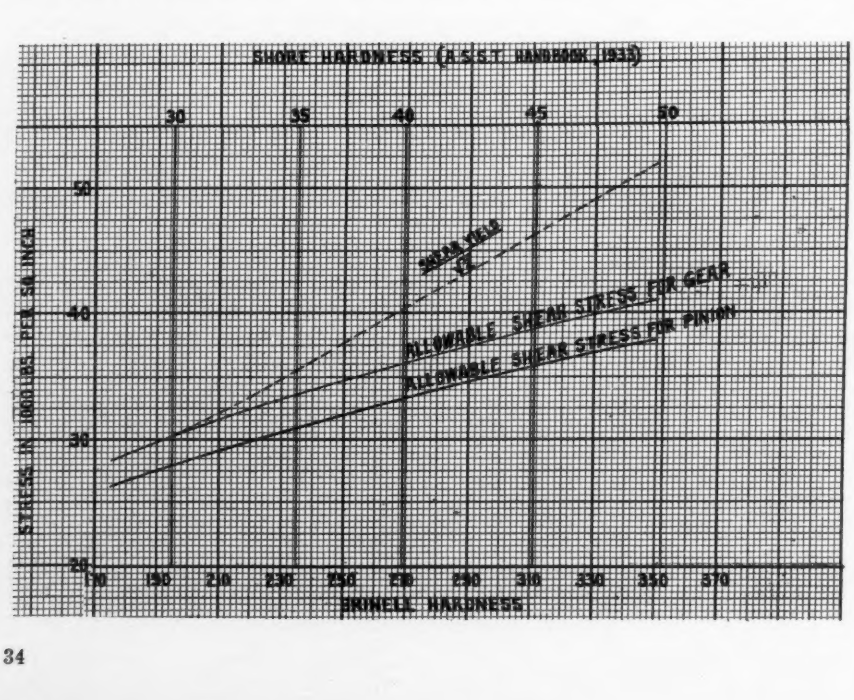


Fig. 27—Allowable shear stress for gears based on shear yield point and endurance limit

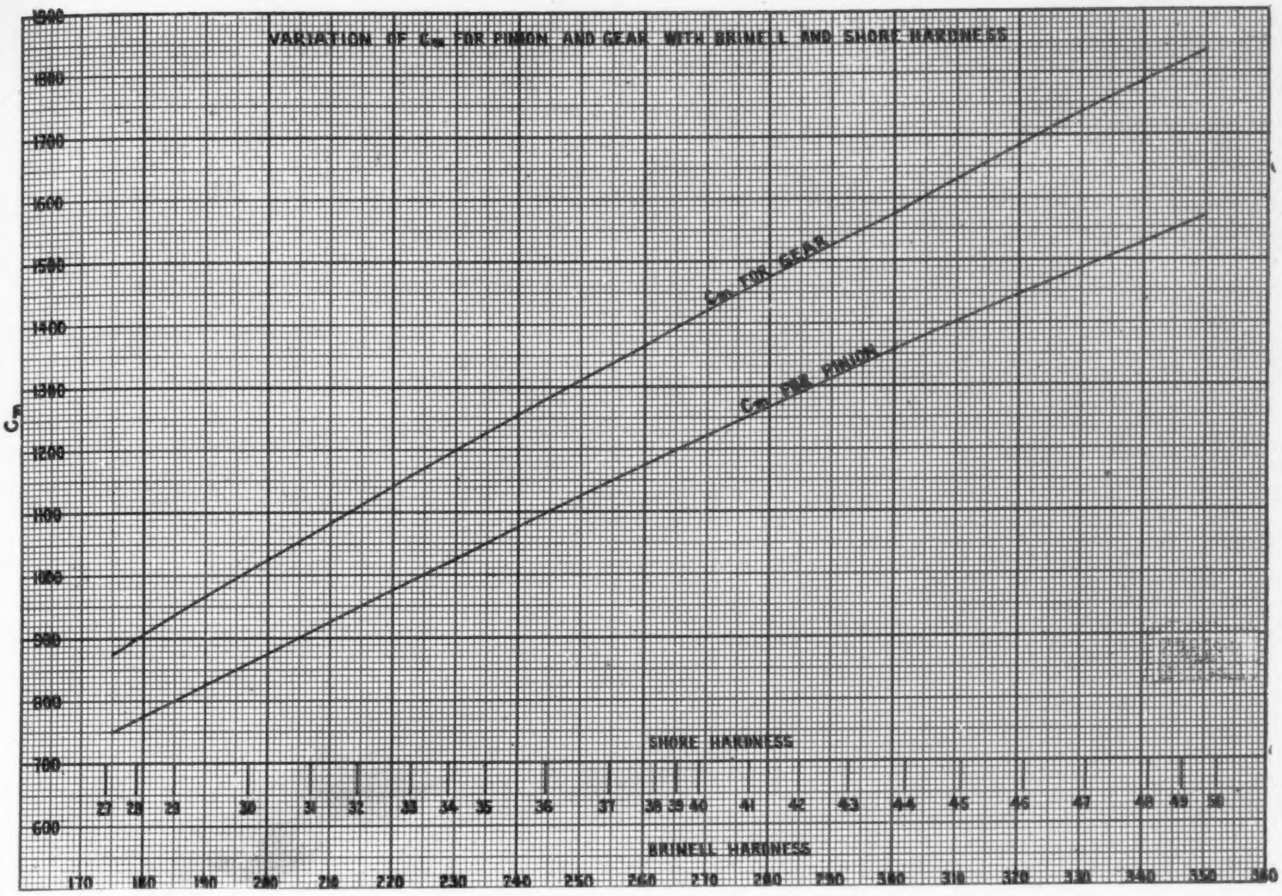


Fig. 28—Material factor varies with Brinell and Shore hardnesses

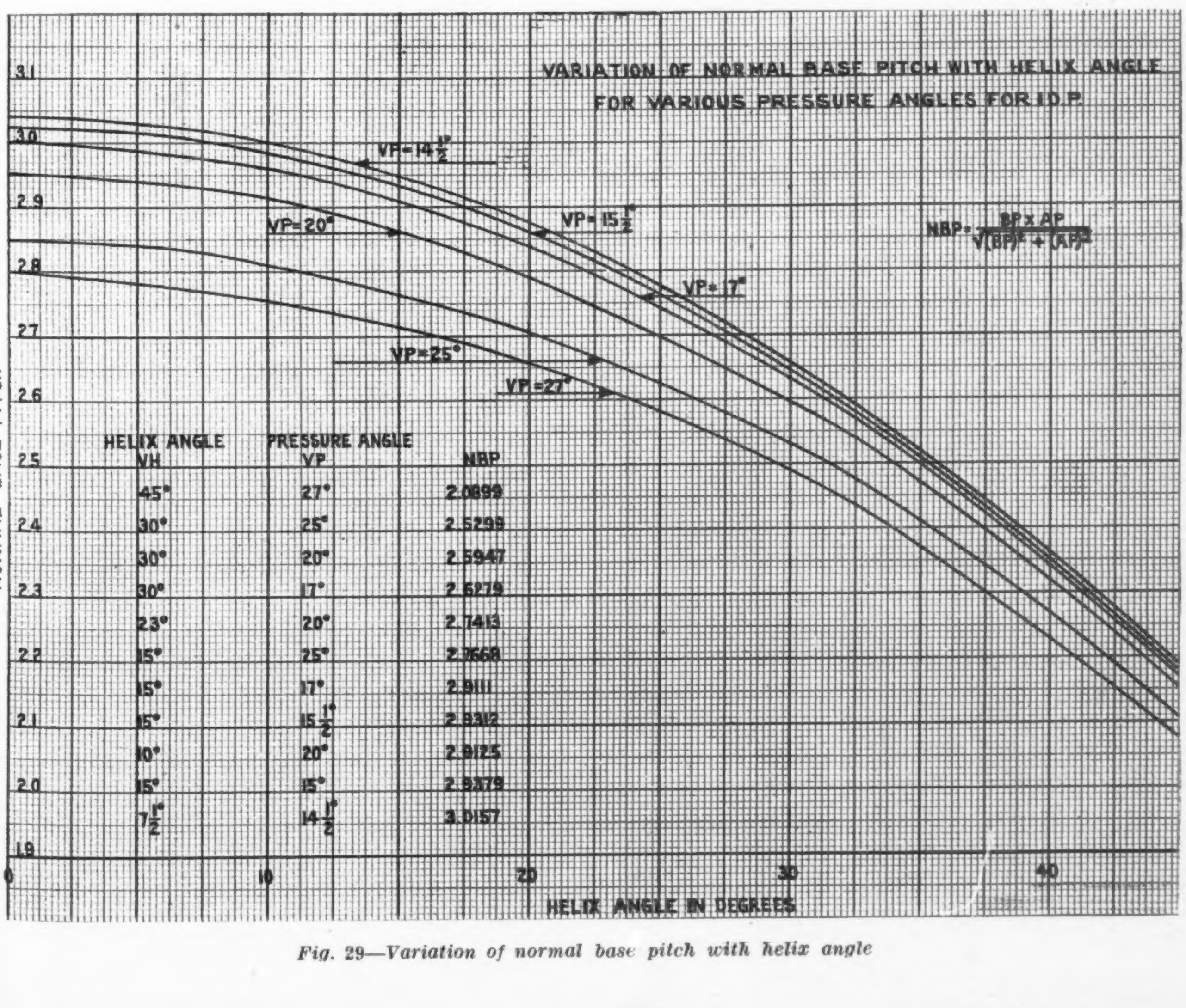


Fig. 29—Variation of normal base pitch with helix angle

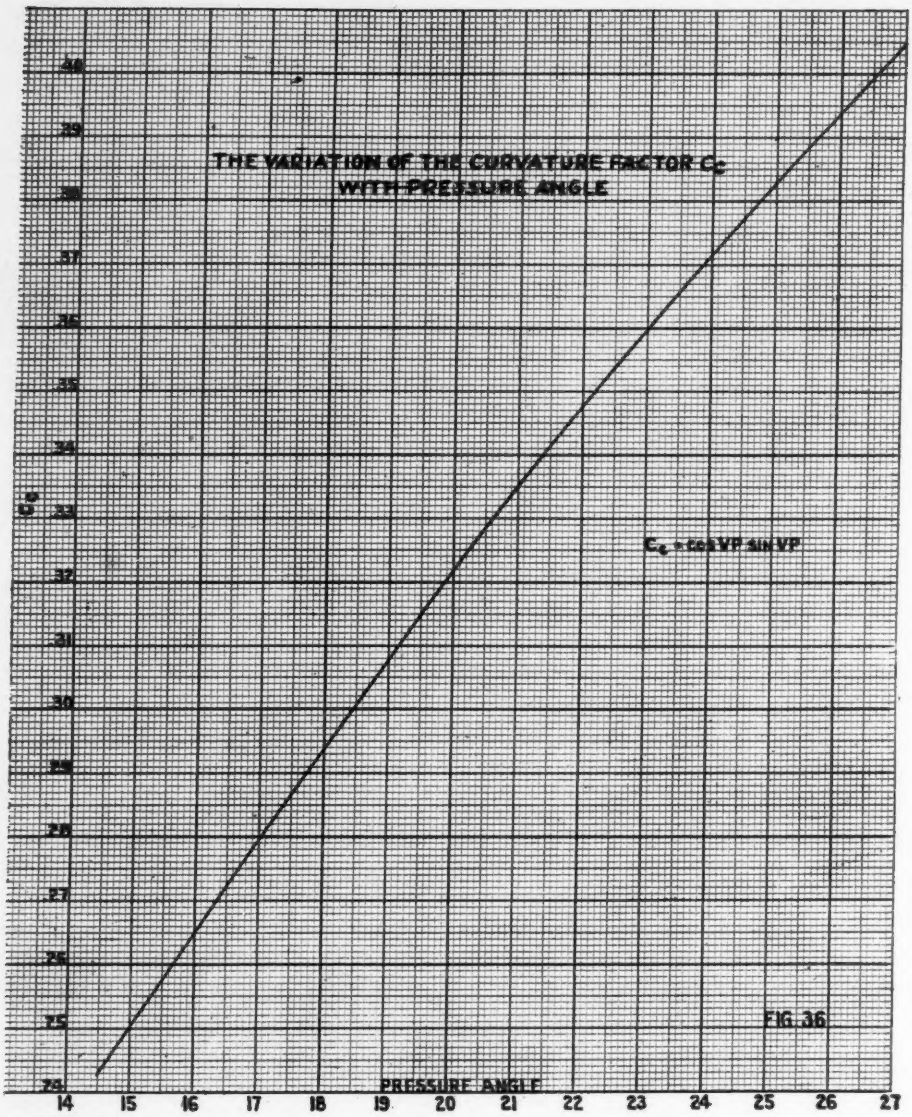


Fig. 30—Curvature factor varies with the pressure angle

stitute as follows

$$P' = TP_n / CL_{min}$$

where

TP_n = total normal pressure on contact line
CL_{min} = minimum length of contact
$r_1 = PD_p \times sin\ VP / \ (2\ cos\ VBH)$
$r_2 = PD_g \times sin\ VP / (2\ cos\ VBH)$

The equation then becomes

$$S_s = 958 \sqrt{\frac{TP_n \times (PD_p + PD_g) cos\ VBH}{CL_{min} \times PD_p \times PD_g \times sin\ VP}}$$

The total pressure in the plane of action equals *TP/cos VP* where *TP* is the total tangential pressure. The total tangential pressure normal to the contact lines is equal to the pressure in the plane of action divided by the cosine of the base helix angle or

$$TP_n = TP / (cos\ VP \times cos\ VBH)$$
$$= TP / (cos\ VNP\ cos\ VH) \quad (Fig. 25)$$

Therefore

$$S_s = 958 \sqrt{\frac{TP \times (PD_p + PD_g)}{CL_{min} PD_p PD_g sin VP cos VP}}$$

Solving for the tangential static tooth pressure

$$TP = (S_s)^2 / 958^2 \times \left[CL_{min} \times PD_p \times PD_g \times sin\ VP\ cos\ VP\right] \div (PD_p + PD_g)$$

The factors entering into the determination of the capacity of herringbone or single helical gears are:

A. *Materials factor*

$$C_m = (S_s)^2 / 958^2$$

where S_s becomes the allowable working shear stress and is equal to the endurance limit S_e or $S_y/\sqrt{f}$ depending on which gives the lower value. Values of C_m are to be taken from *Fig.* 28.

B. *Contact Length factor*

$$C_q = CL_{min}/FW = [K \times (FW \times LA) / NBP] \div FW$$
$$= (K \times LA) / NBP$$

$C_q = 0.9LA/NBP$ for general helical gears

$C_q = 0.95LA/NBP$ for standardized speed reducers when LA is taken for a ratio of 8:1

Values of LA are to be taken from *Figs.* 13-18 (M.D. June, pp. 56,57, 58) and NBP from *Fig.* 29.

C. *Curvature factor*

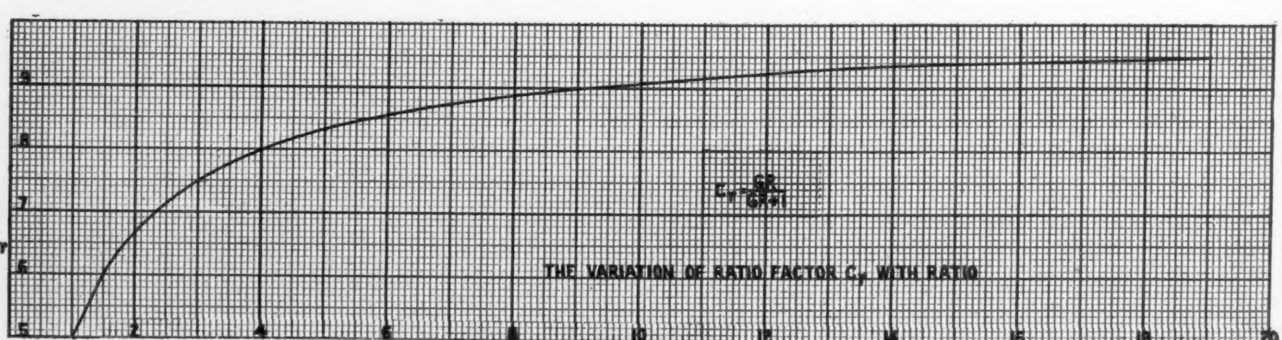


Fig. 31—The variation of the ratio factor with the reduction ratio

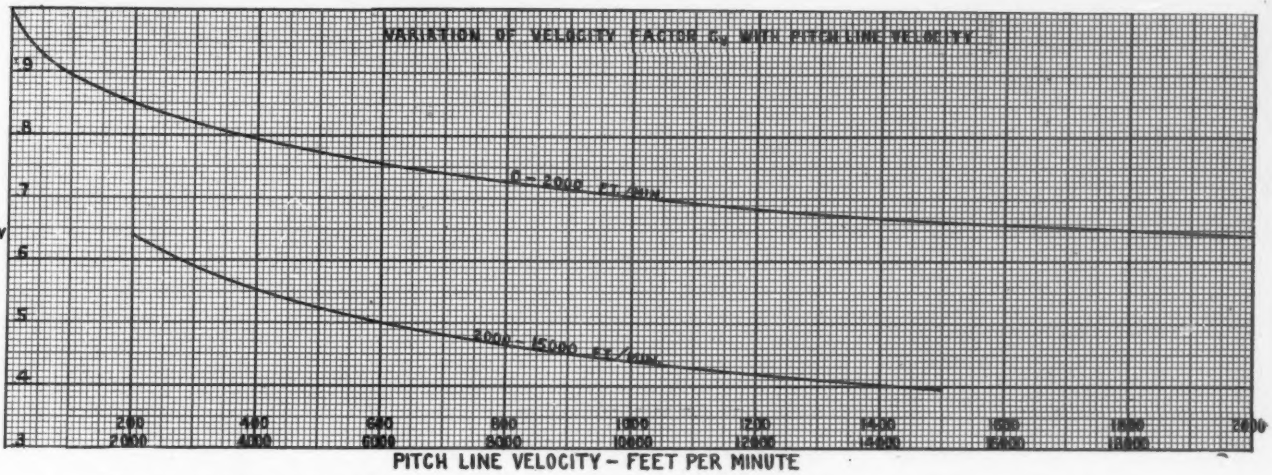


Fig. 32—Values of the velocity factor for various conditions may be taken from this curve

$$C_c = \sin VP \cos VP$$

Value of C_c are to be taken from *Fig.* 30

D. *Ratio factor*

$$C_r = GR/(GR+1)$$

where GR is the gear ratio. This value is obtained by reducing the quantity $(PD_p \times PD_g)/(PD_p + PD_g)$ to $(GR \times PD_p)/(GR + 1)$

Values of C_r are to be taken from *Fig.* 31.

E. *Pitch diameter, PD_p*

F. *Face width, FW*

G. *Velocity factor, $C_v = 78/(78 + \sqrt{V})$*

Values of C_v are to be taken *Fig.* 32.

H. *Inbuilt factor, C_i*

Values of C_i are to taken from *Fig.* 33.

Using these factors, the equation for allowable unit pressure becomes:

$$TP/FW = C_m \times C_q \times C_c \times C_r \times PD_p \times C_v \times C_i$$

Regardless of whether the gear or pinion is the weaker, the capacity should be figured using the pinion diameter and ratio factor. The contact stress is a mutual stress, having the same value in gear and pinion regardless of ratio. There is no need to compute the capacity of gear and pinion separately because their respective C_m values indicate beforehand which is the weaker The computation is based on the pinion diameter merely for convenience.

A study of tooth form combinations in everyday use shows a variation of less than plus or minus 10 per cent for the product of C_q, C_c. The minimum is about 0.4 for the conditions examined.

The hardness differences shown by the curves agree quite closely with present day practice and may be used as a guide. It is advantageous to have the pinion even harder than indicated, because of the beneficial effects on the gear. When the hardness spread is less than shown by the relation of the two curves, the C_m values must be taken from the pinion curve.

It might appear logical to use a contact factor to take care of the greater number of contacts on the pinion in accordance with ratio. Experience indicates that there is no necessity of introducing such a complication if the difference in hardness is great enough to take care of the

(Concluded on Page 44)

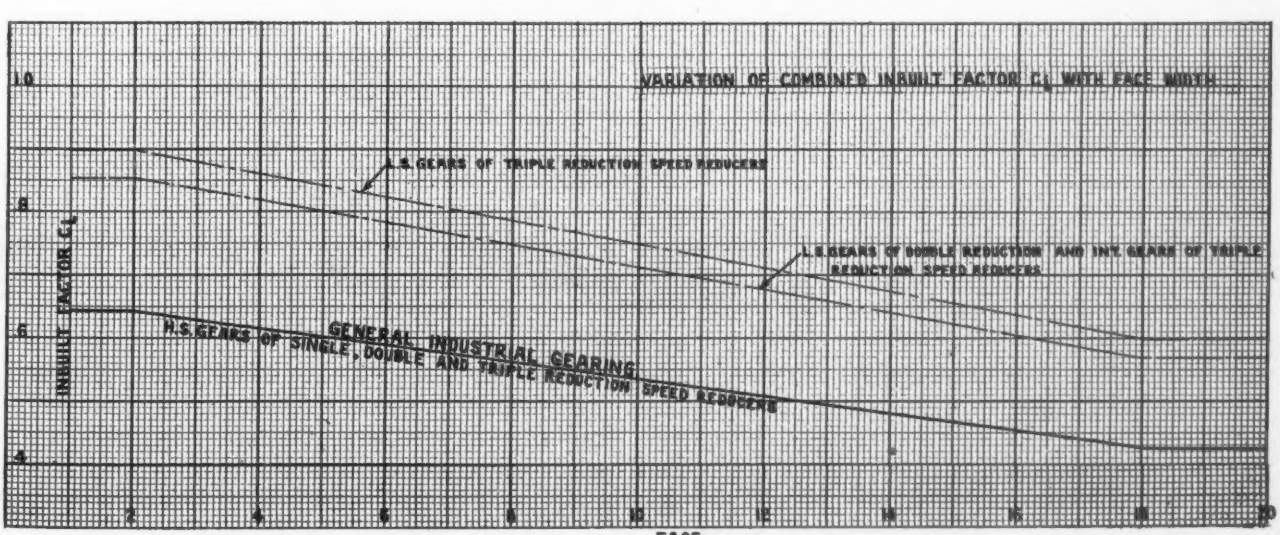
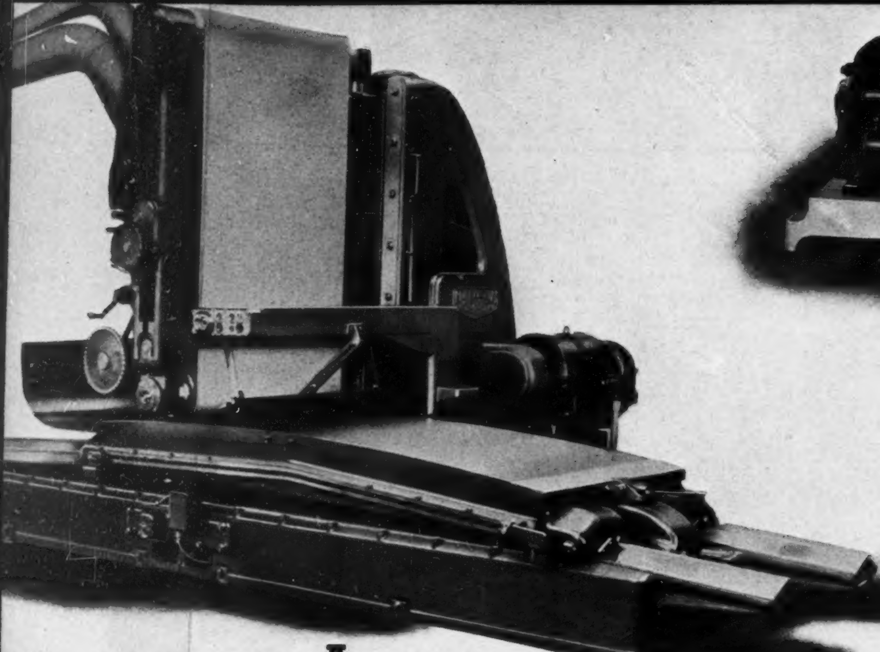


Fig. 33—Values of the combined inbuilt factor as it varies with face width are obtainable from this curve



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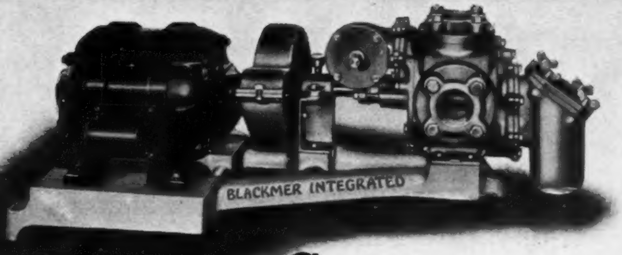

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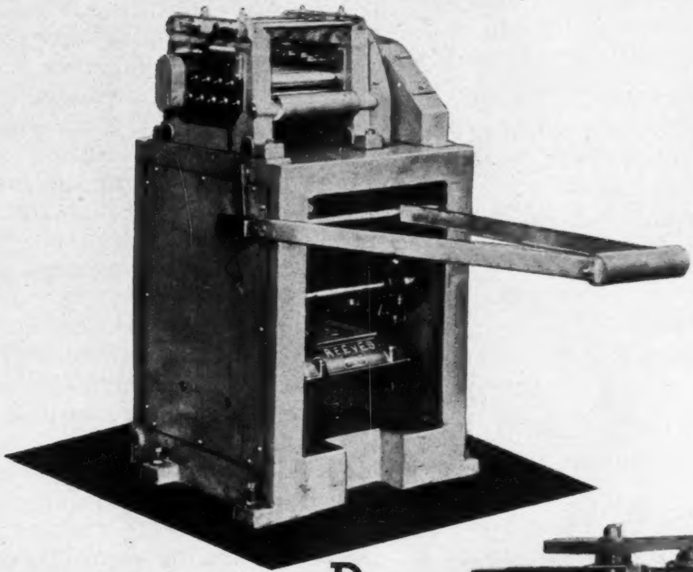

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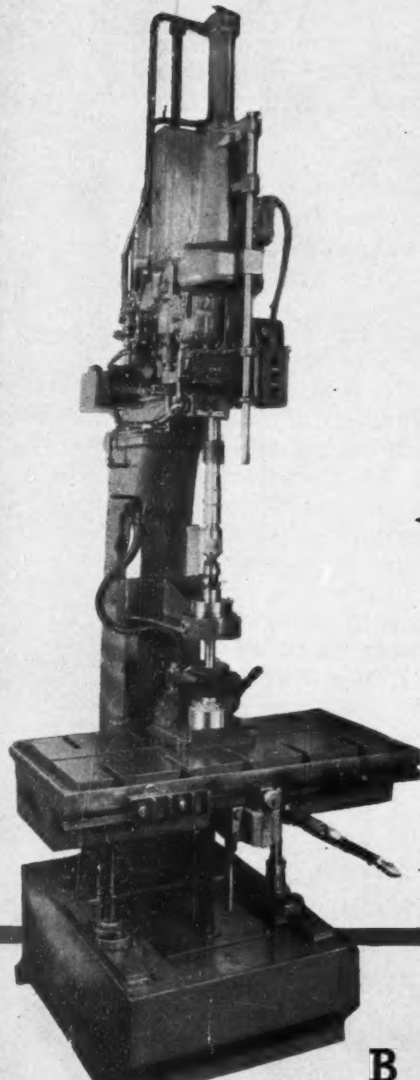

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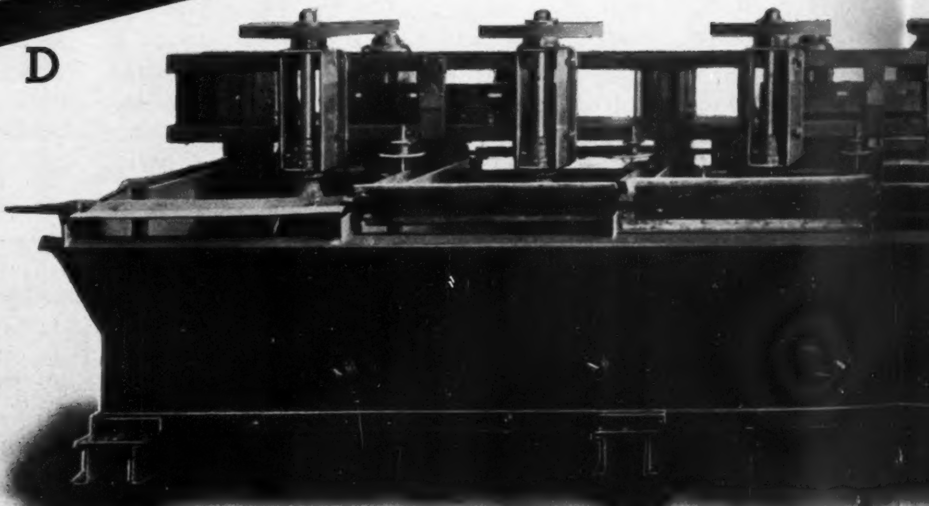

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(A) Main arm of abrasive belt sheet grinder, carrying the belt, driving and idler rolls, is adjustable as a unit for different thicknesses of material and for setting to the cut. These grinders have been designed and built by Mattison Machine Works with the necessary strength to insure rigidity and freedom from chatter and vibration.

(B) Electrically controlled stroke counter on small hydraulically reciprocated honer of Barnes Drill Co. not only predetermines the number of strokes, but provides for automatic approach into the work, automatic assumption of the honing or working stroke, and automatic spindle return when the predetermined number of strokes has been made.

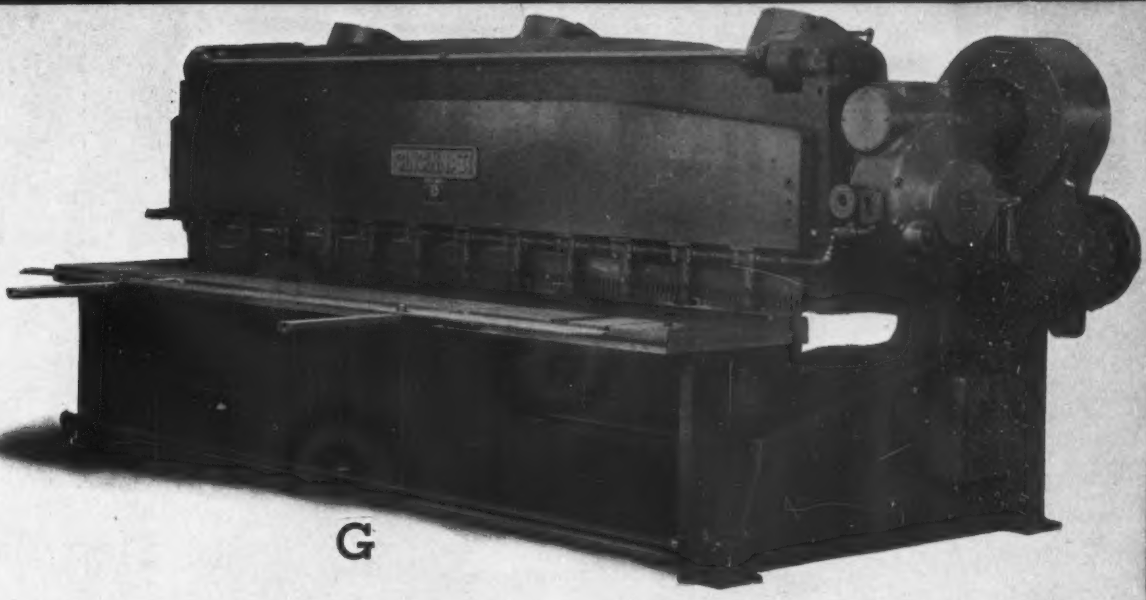
(C) Three-point support principle adapted from aeronautic design is used on integrated pumps of Blackmer Pump Co. Drive is totally enclosed. Full size bypass obviates unbalanced load on pump rotor. Base is of channel sections.

(D) Roll feeding straighteners are individually motor driven. Constant speed motor furnishes power to a mechanical variable speed unit. An automatic control on this machine, built by Waterbury Farrel Foundry & Machine Co., shuts off power when press stops.

(E) Traffic line marking machine of Simons Paint Spray Brush Co. is a complete power plant having an aircooled foot starter valve gasoline engine, V-belted to the air reservoir, air passes throug control pressure on the paint a to the spraying nozzle. One ar handling ease.

(F) Single unit ball bearing machine for treating go practically eliminated vibration alignment. The machine, buil Co., has been redesigned to ma struction. Clear visibility of ev has been provided in the desi erate the individual heads.

E



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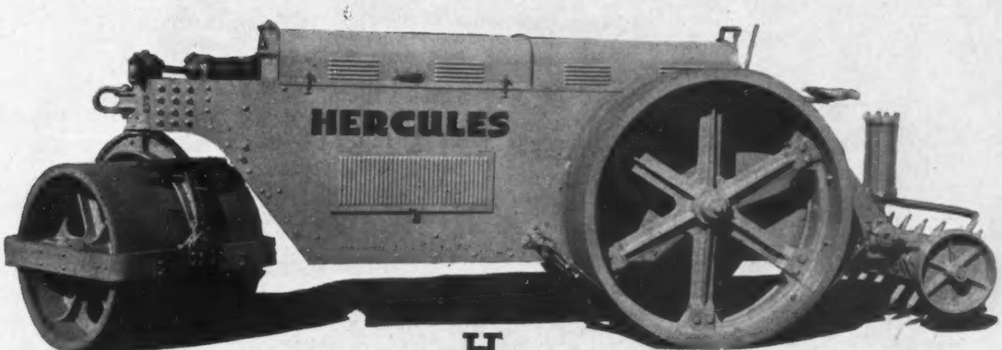

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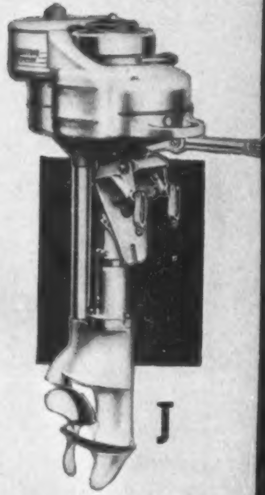

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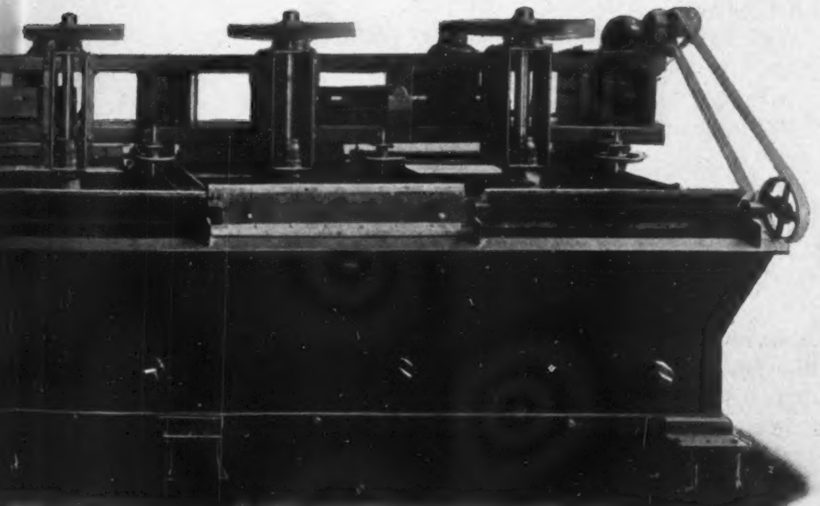

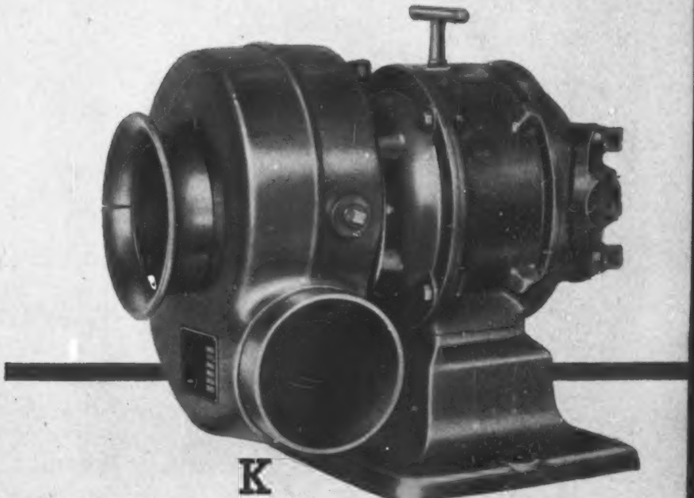

K

r valve in the magneto type
to the compressor. From the
through regulator valves that
oint and a second line passes
one and ball bearings increase

aring assembly in flotation
g gold and other ores has
ation, while it insures shaft
, built by Denver Equipment
o make use of all steel con-
of every cell by the operator
edesign. V-belt drives op-

(G) All steel plate construction insures strength and accuracy in new shear of Cincinnati Shaper Co. Hydraulic holddowns deliver a heavy uniform pressure, preventing sheet slippage. Drive is V-belt from motor to flywheel, then through a silent worm and wheel reduction unit to the main-shaft. Worm is mounted on antifriction bearings.

(H) Ten-ton road roller has cast steel steering arm to which is connected a hydraulic piston for hydraulic power steering. Appearance has been given special attention in the design. A cast steel spider forms the spokes in the two-piece center rear roll, and separate semi-steel rims of any width may be mounted thereon.

(J) A spiral bevel gear drive is employed on new outboard motor of Outboard Motors Corp. Units are floated in rubber mountings which insulate both the steering handle and the boat. The rotary disk valve is an integral part of the crankshaft, thus cannot get out of time.

(K) Cast aluminum is used throughout in watertight ventilating set built by Diehl Mfg. Co. The motor is totally enclosed and is equipped with ball bearings. Blower has efficient impeller made of corrosion resisting material. Handle for carrying set from place to place is mounted in motor frame. Internal connection block and watertight bushing are provided.

MACHINE DESIGN

Best Design Is the One that Meets the Wishes of Most Customers

THE customer is coming into his own! A recent survey of the machinery building industry indicates that at this time more than ever before the specific requirements of customers rather than the whims of shop superintendents or sales managers are influencing designers in the development of their machines.

This same survey reveals that there are few companies who are not spending money on redesign at the present time. The opportunity to take advantage of improved business possibilities by collaborating with customers is at hand.

Such a condition has held true in the past only with regard to special machinery. On standard machines it always has been a difficult matter to convince the manufacturer, and particularly the designer, that a change in design would be advantageous. Designers, due to pressure that is apt to be exerted on them from all quarters within the company—from executives to shop foremen—necessarily have had to take a somewhat unconciliatory stand toward proposed changes. Otherwise they never would cease to be trying to meet the suggestions, often immature, of everyone in the plant.

This attitude has gone into the discard as far as the customer is concerned, together with many other things, during the depression. The designer or company that is alert to the wishes or suggestions of clients is in an excellent position to benefit. Independence and bull-headedness have no place in present-day manufacturing.

• • •

Machine Noise Is Doomed!

ELIMINATION of noise in machinery has been stressed repeatedly in these columns. Indicative of the increasing trend toward this objective is the establishment and popularizing of a standard unit of noise measurement, the decibel, and the recent building of a silence chamber in one of our largest manufacturing plants. Formerly such chambers were used primarily for testing musical tones. Their use by manufacturing plants augurs well for the eventual demise of noisy and nerve-wracking machines.

Another factor that will assist greatly in efforts to reduce noise is the development of noise meters and noise analyzers. Not only are these being used to test machines in specially built chambers, but portable models are being employed on transportation equipment and other machinery in service.

Rubber, springs and other types of insulation also are being given prominent attention. The day is not distant when "one will be able to hear one's self speak"—regardless of the location.

MEN OF MACHINES

HONOR has again been bestowed upon William E. Sykes. To him recently was awarded the Edward Longstreth medal by Franklin Institute "in consideration of the design and development of the Sykes gear generator." He was born in England in 1883 and was graduated from the Huddersfield technical college of Leeds university.

In 1911, when chief engineer and factory manager of the Power Plant Co. in England, he commenced the design and development of gear generators. Simultaneously he has developed the Sykes gear which has continuous herringbone teeth.

Mr. Sykes also is the designer of various other unique machines employed in gear manufacture. He is affiliated with Farrel-Birmingham Co. in Buffalo.

WILLIAM E. SYKES

• • •

MELVIN L. ENGER

TO HEAD the college of engineering and engineering experiment station of the University of Illinois, its trustees have chosen an alumnus, Prof. Melvin L. Enger. A member of the university staff since 1907, his field of specialization has been engineering mechanics and hydraulics.

Prof. Enger was born in Decorah, Ia., in 1881. After pursuing his earlier studies at the University of Minnesota he entered the University of Illinois to finish his undergraduate work. Obtaining two degrees in 1906 and 1911 respectively, he was the recipient of the degree of master of science in 1916.

On the retirement of Prof. A. N. Talbot in 1926 he became head of the department of theoretical and applied mechanics. He is in charge of research investigations in engineering materials.

• • •

AS EARLY as 1897 Dr. D. S. Jacobus, new president of the American Welding society, made experiments with the oxyacetylene torch in cutting thin steel. In addition to his pioneering work in that field, he is considered an authority on steam engineering. Dr. Jacobus has served as president of the American Society of Refrigerating Engineers and as president of the American Society of Mechanical Engineers.

Born in Ridgefield, N. J., he was graduated from Stevens Institute of Technology in 1884. From that time until 1906 he taught experimental mechanics and engineering of physics there. In the latter year he was awarded the degree of doctor of engineering. Subsequently he joined the Babcock & Wilcox Co., New York, as advisory engineer, and now heads the company's engi-

D. S. JACOBUS

eering department.

Dr. Jacobus has been a member of the board of directors of the Welding society for several years and also holds memberships in numerous other technical organizations.

* * *

MARK LOVEL NICHOLS, head of the department of agricultural engineering, Alabama Polytechnic institute, Auburn, Ala., has been awarded the 1934 Cyrus Hall McCormick medal by the American Society of Agricultural Engineers.

* * *

WILLIAM EDWARD BOEING has been awarded the 1934 Daniel Guggenheim medal for successful pioneering and achievement in aircraft manufacturing and air transport.

* * *

WILLIAM E. UMSTATTD is the new president of Timken Roller Bearing Co., Canton, O. H. H. TIMKEN, who recently resigned from that office, remains as chairman of the board.

* * *

CHARLES F. KETTERING (M. D., June, 1930, p. 56), vice president of General Motors, was the recipient of the honorary degree of doctor of engineering from the University of Detroit. He also delivered the commencement address.

* * *

W. J. DONALD is the new director of the National Electrical Manufacturers association, succeeding A. W. BERRESFORD, resigned.

* * *

E. G. K. ANDERSON, for many years chief engineer of Appleton Electric Co., has become associated with the newly formed Simplet Electric Co., Chicago.

* * *

PAUL WARE recently was appointed chief engineer of Emerson Radio & Phonograph Corp. He was the designer of the "Ware neutrodyne" sets which were introduced during the early days of broadcasting.

* * *

W. E. ENGLAND recently was appointed chief engineer of Ohio Rubber Co., Willoughby, O. For nine years he was chief engineer of F. B. Stearns Co., and more recently was head of Auburn's experimental department.

* * *

ALFRED V. DEFOREST, widely known research engineer of Bridgeport, Conn., joins the staff of Massachusetts Institute of Technology in the autumn as associate professor of mechanical engineering.

* * *

R. E. WILKINS recently was appointed manager of the automotive engineering department and assistant manager of the technical department of the Standard Oil Co. of Indiana. H. J. SALADIN has been made manager of the technical department.

* * *

EUGENE G. GRACE, president, Bethlehem Steel Corp., has been awarded the Gary medal for outstanding achievement in the iron and steel industry.

Should Prepare Patents with Care

(Concluded from Page 27)

patented; either by fixing thereon the *word patent; together with the number* of the patent; or when, from the character of the article, this cannot be done, by fixing to it, or to the package wherein one or more of them is inclosed, a label containing the like notice: and in any suit for infringement by the party failing so to mark, no damages shall be recovered by the plaintiff, except on proof that the defendant was duly notified of the infringement and continued, after such notice, to make, use, or vend the article so patented."

Early Examination Is Possible

Sometimes, commercial situations make it unprofitable to delay the exploitation of an invention until the patent is granted in its regular course. In such cases, if the invention is put on the market while the patent application is pending, the patent office may make the patent application "special" and advance for early examination upon a proper showing made by the inventor to the effect that the inventor's device is being copied by competitors and that he will be irreparably damaged unless the patent is granted immediately.

The words "Patent Pending," or "Patent Applied For" stamped or otherwise placed upon the inventor's device have no legal effect. They merely give information that an application has been filed in the patent office and that a patent may issue in due course. This may or may not scare away competitors. If it requires a great deal of money to tool up the plant to make the inventor's device, competitors may be indisposed to spend the money knowing that they may be enjoined upon the issuance of the patent. If it takes very little money, the competitors may go ahead notwithstanding the words "Patent Pending." It should be noted, however, that such practice is made unfair competition by most of the NRA codes. While the codes for the several industries vary in language the following is given as an example:

"Appropriating or attempting to appropriate ideas, sketches, designs or drawings originated and owned by another in the industry without the owner's consent," is made unfair competition.

Articles on this and allied subjects published in previous issues of MACHINE DESIGN include:
"Careful Research Obviates Patent Pitfalls," March, 1933, p. 25; "Value of Patent Depends upon Efficient Claims," Aug., 1933, p. 13; and "Protect Your Right to a Valid Patent!" Nov., 1933, p. 24, all by George V. Woodling.
"Outstanding Patent Book Revised," Feb., 1934, p. 23.
"Patents, Inventions and Design Ingenuity Revitalized Under New Deal" (an editorial), March, 1934, p. 36.
"Patent Contracts Insure Legal Protection," by George V. Woodling, April, 1934, p. 29.

TOPICS

REPORTS from the Chicago World's Fair indicate that the technical man is finding much to interest him again this year. The electrical building is one division where unique devices are being shown. In this large exhibit for example, is located a metal detector that is being introduced in police departments for the location of concealed weapons on criminals. This machine will signal if the person standing before it has any metal concealed about his person. Industrial plants will find this device useful in locating foreign metal particles in their products. Another development demonstrated is a patented metal band designed to be installed in homes. It runs around the wall of the room, being positioned at the top of the baseboard. The electric light plug may be inserted in this strip at any desired point in its entire length.

* * *

Tires made entirely of a synthetic product known as DuPrene have incited much comment. In announcing the development the Dayton, O., company by which the new tires have been built declares that this synthetic rubber does not age appreciably or deteriorate when exposed to the action of the elements or rubber solvents.

* * *

Judges at the recent exhibition of machine art which attracted thousands of visitors to the Museum of Modern Art in New York, selected the three most "beautiful" objects. Their first choice was a section of a large spring made by American Steel & Wire Co.; second, an outboard propeller produced by Aluminum Co. of America; third, an SKF ball bearing.

* * *

When announcement was made that drivers in this year's 500 mile Memorial Day racing classic at Indianapolis would be limited to 45 gallons of fuel the research laboratories of the Standard Oil Co. of New Jersey produced a superpower fuel having a high energy content per gallon. Early tests revealed that the new gasoline reduced fuel consumption by about 12 per cent. In the race Wild Bill Cummings' big 225 cubic inch Miller, which carried him to victory, consumed only 35½ gallons of the 45 permitted. The development is particularly interesting because some drivers felt that the large type racing cars could not finish the race on the new fuel allotment.

* * *

Evidence now is at hand to show progress on various industrial fronts since the inauguration of the national recovery program. Recent figures gathered from a number of sources reveal that for the first quarter of 1934, 210 industrial corporations which aggregated a deficit of $23,000,000 for the same period last year, report a profit of $98,000,000. Sixty-four miscellaneous companies covering a wide range of business activities report a profit of $27,151,396. Their deficit for the first quarter of 1933 totaled $29,500,000.

* * *

An experimental diesel embodying arc welded cylinders is being built by F. B. Stearns of Cleveland. Since both weight and space are at a premium in marine type engines, castings were discarded and arc welded steel employed for the cylinders and other parts. Design and arc welding of these cylinders comprised a difficult problem since some sixteen pieces are required for each and the tolerance on the finished work is exceedingly small. The inner sleeve is of case hardened steel and the outer portions of mild steel.

* * *

Word comes from Rome that Enrico Fermi, a member of the Italian Lincei academy, has discovered a new element known as No. 93. It was created artificially from uranium and is said to be radioactive, with chemical properties analogous to those of manganese. The new element also is claimed to be the hardest known to man.

Determining Capacity of Gearing

(*Concluded from Page 37*)

maximum ratio considered.

The minimum C_m value of 875 shown for the gear is to be used for annealed or untreated gear steels. All hardnesses must be obtained by accepted heat treating methods. Hardness is usually specified as a range. The minimum should always be used, never the average. The manufacturing department is inclined to work to the low limit in order to facilitate production. Thus, if the gear drawings are marked 225-260 brinell, the C_m value to be used is 1170. If the pinion of this set is specified 250-285 brinell, the C_m value for pinion is 1125. The gears

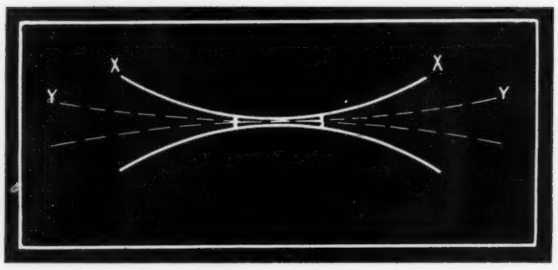

Fig. 34—Analogy using the relative radii of curvature of two contacting teeth to show increase of contact by the lubricant (exaggerated)

must be computed for the lower of the two values, 1125.

Horsepower Equations

By substitution

$$(a)\,H.P. = \frac{(C_m \times C_q \times C_c \times C_r \times PD_p^2 \times C_v \times C_i \times FW \times RPM)}{126{,}000}$$

This is the equation based on average contact length or

$$C_q = K \times LA/NBP$$

(b) For general application where proportions are not definitely known or where more exact results are not required, the product of C_c, C_q can be assumed as 0.4.

$$H.P. = \frac{(C_m \times C_r \times PD_p^2 \times C_v \times C_i \times FW \times RPM)}{315{,}000}$$

The above equation is obtained by taking the contact for an 8 to 1 ratio with 100 teeth in the gear train and $CL_{min} = 0.9.\,CL_{ave}$.

Where detailed studies are warranted, as in technical comparisons the actual CL_{min} values can be computed by the method shown in *Figs.* 19, 20 and 21 (M. D. June, p. 59) and the equation then becomes

$$H.P. = \frac{(C_m \times CL_{min} \times C_c \times C_r \times PD_p^2 \times C_v \times C_i \times RPM)}{126{,}000}$$

Adjusting for Nature of Service

It is a rather difficult matter to interpret the effect of shock loading on the dynamic stress. A suddenly applied load produces stresses of twice the intensity of those obtained with a gradually applied one. Allowance has been made for 100 per cent overload in the equation so that if the exact value of the shock load were known it would be possible to compute the size of gears from this load without any additional factors. The duty expected of a set of gears is seldom if ever expressed in that manner. The average load or motor horsepower is more frequently specified. The peaks then depend on the energy stored in the rotor and the other rotating elements. Where there is a possibility of the load jamming, the instantaneous loads can reach tremendous values. The frequency and duration of the peak loads should be considered. If the maximum occurs once every five minutes with a duration of 1/10 second, the gears carry the maximum load only 1/3000 of the time. If the gears are lightly loaded the remainder of the period there is plenty of justification for considering this as intermittent service, even though the machines operate 24 hours a day. More good judgment is necessary in determining the load to design for than in applying formula.

A set of service factors suitable for average conditions are shown in Table I. After the rating has been computed by the equation for eight hour uniform loading, it should be divided by the factor shown for the particular type of service to which the gears are to be applied.

Lubrication Must Be Considered

The surface areas which develop to resist the pressure are due chiefly to elastic deformation of the material, but the cushioning effect of the lubricant is a contributing factor. *Fig.* 34 represents two teeth in contact. The amount of contact between them can be measured by the reciprocal of the relative curvatures. Due to the dynamical characteristic of the lubricating medium, the film is probably thicker at the end of the contact band where pressure is lower than at the center where pressure is a maximum. The effect is to produce the greater curvature illustrated by "*y-y*" as compared to "*x-x*". There is reason to believe that with heavier lubricants this beneficial effect is more marked. Certain lubricants which have in their makeup materials of a solid nature, such as lead, talc, etc., give the gears greater resistance to failure by pitting. This is not said with the idea of advocating general adoption of such lubricants. They have disadvantages which in a good many cases offset the advantage of greater load carrying capacity. There are, however, applications where they are used to advantage. Likewise, semifluid compounds made up exclusively of petroleum products have their logical applications. Some arbitrarily assigned measure must be given to these lubricants in considering their effect on the load carrying capacity.

The lubrication of geared drives may be divided into four general classes depending upon the method of application. These are as follows:

TABLE I

Helical and Herringbone Speed Reducer Service Factors

	8-10 Hour Service	24 Hour Service	Intermittent Service 3 Hrs./Day
Uniform Load	1.0	1.25	0.8
Light Shock Load	1.25	1.5	1.0
Medium Shock Load	1.5	1.75	1.25
Heavy Shock Load	1.75	2.0	1.5

Open gears where the lubricant is painted onto the teeth from time to time. This method can be used only for extremely slow applications and is so unreliable, and depends to such a large extent on the personal interest of the operator that it is rapidly losing favor

Splash lubrication in which gears dip into the lubri-

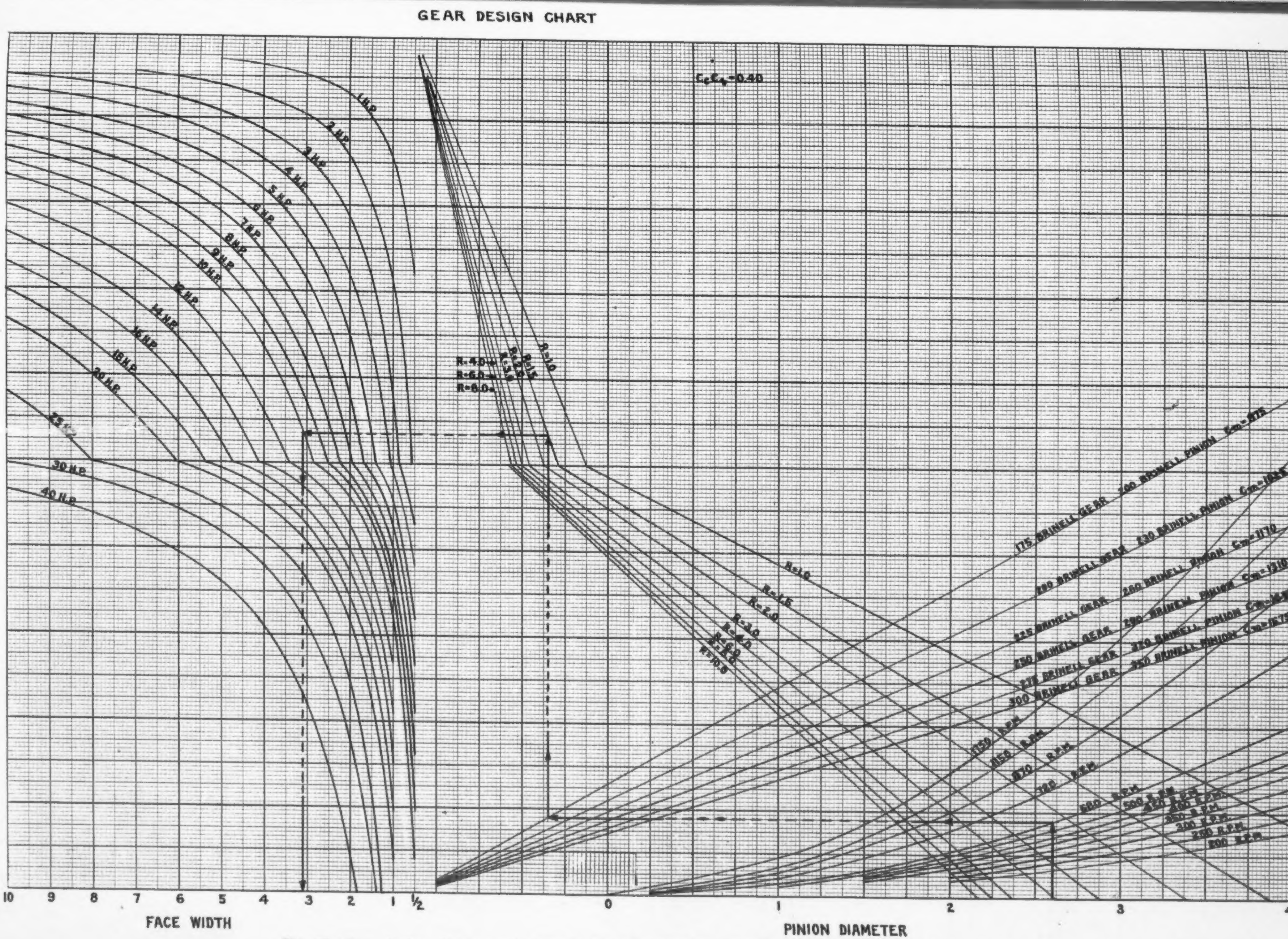

Fig. 35—Gear design chart permits face width determination with loads from one to forty horsepower

cant retaining a sufficient quantity to satisfy the gear mesh, and throwing a supply into a trough from which the bearings are fed. In this case the same lubricant is used for both the gear mesh and the bearing and, therefore, it will be a compromise lubricant

Pressure lubrication, in which the gear lubricant is introduced at the mesh point by sprays and the bearings are pressure lubricated. Here again the lubricant must suit the requirements of both gears and bearings, but since this class consists largely of high speed drives, lubricating conditions on the gears are excellent, and the comparatively thick oil film separating the teeth gives ample protection

Separate lubrication, in which the gear dips into a heavy compound selected to handle the gear mesh in the best manner possible and the bearings are lubricated by an oil intended to make them operate at low temperatures.

Where mineral oils are used the viscosity and film strength should increase as the pressure per inch of face increases. The contact band is wider and, therefore, the amount of work which the lubricant has to do is greater. The following rules will be found to follow good practice closely for normal operating room temperatures of 40 degrees Fahr. to 100 degrees Fahr.

Pitch Line Velocity in feet per minute	Minimum Viscosity in Saybolt Sec. at 100 degrees Fahr. per 100 pounds pressure per inch of face	Minimum Total Viscosity
Up to 2,000	100 Sec.	500 Sec.
2,000 to 3,000	75	500
3,000 and up	50	350

The maximum oil viscosity should not exceed the minimum values by more than 50 per cent. For room temperatures 100 to 140 degrees viscosities should be 40 per cent to 50 per cent higher, and for temperatures from 0 to 40 degrees they should be about two-thirds of the values required for normal operation.

For oil lubrication and where the above specifications are met the lubrication factor should be taken as 1, that is ratings can be used as computed from the equation. This applies for both pressure and splash lubrication.

Where the use of a compound of the gear shield type is logical, such as for slow speed, heavy shock loads, the load carrying capacity can be increased 25 per cent.

Where heavy E. P. lubricants, such as lead soaps of approved types are proper, the capacity may be computed as 33 per cent greater than in the case of mineral oils.

No definite rules can be laid down for applications where the lubricant is painted onto the gears since so much is dependent on the personal element.

Face Width Limited

There are certain limitations to the face width which can be used for any particular combination of pinion diameter and loading. It is natural to expect that the pinion shaft will deflect between its supports. If this deflection is excessive there will result a difference in the intensity of loading over the face. The difference in bearing at the center and ends of the face determines the stress concentration. It is the maximum and not the average contact stress which is important. Consequently the allowable deflection under full load must be limited by properly chosen relations of face width and diameter.

For ordinary applications the face width should not exceed two times the diameter of the pinion and it is better to aim for 1½ diameters. There are a good many applications of helical and herringbone gearing where 2½ diameters has proven satisfactory, but this should not be used except where there is an established precedent.

As a general rule, the higher the hardness (and consequently the loading) the smaller the allowable deflection.

The position of the bearings with respect to the pinion, the stiffness of the shaft and the possible reaction from gears carried on the pinion shaft may influence the deflection to the extent that the empirical relations of diameter and face must be modified. Where there is a question of doubt a complete analysis should be made by determining the actual deflection by graphical integration.

In single helical gears the tipping moment resulting from the helical thrust reaction must be limited in order to prevent load concentration and noise. This can be accomplished by choice of helical angle, shaft supports, etc.

Discusses Contact Stress Variation

It is evident that it is possible to obtain gears having considerable variation in the total contact and it would seem possible to obtain gears having theoretically no variation in the total contact. A method of obtaining gears having an integral number of teeth in contact, or constant lengths of tooth contacts has been developed by A. B. Cox and is covered by U. S. Patent No. 1,525,642. It can readily be seen that increasing the addendum of gear and pinion increases the contact length by increasing the line of action LA in the plane of rotation; and it is obvious that such modifications can be adopted as would make LA a multiple of the base pitch. Likewise, it can be seen that a face width can be chosen which will be a multiple of the axial pitch. If either of these two conditons are fulfilled there will be no variation in the total contact CL. The mathematics of the theory in its evolution is quite ingenious, but there are several considerations which make the actual attainment of the theoretical properties difficult. For most applications where quietness is essential the contact conditons should be checked after the gears are designed to make sure that there is no excessive variation.

The contact variation will be small if the final check shows that either LA/BP or FW/AP is fairly close to a whole number. It does not follow that if neither of these quantities meet this requirement that the contact variation will be excessive because the two can produce good contact by virtue of their complimentary action.

Determining Gear Proportions

The determination of the proportions required to transmit a given horsepower has always been a matter of trial and error. The reason for this is apparent when we consider that there are an infinite number of combinations of face width and diameter which will carry that load. The problem is especially complex because of the fact that the velocity factor depends on the diameter which has not yet been ascertainedd. Usually the speed of the driver and the ratio are specified. Sometimes the centers are known, in which case the diameter of the pinion is equal to $(2 \times centers)/(GR + 1)$. The face width is in that case the unknown quantity and must be solved for. The inbuilt factor C_i depends on the face width so that the selection is complicated still further.

To simplify the design of herringbone or helical gears,

GEAR DESIGN CHART

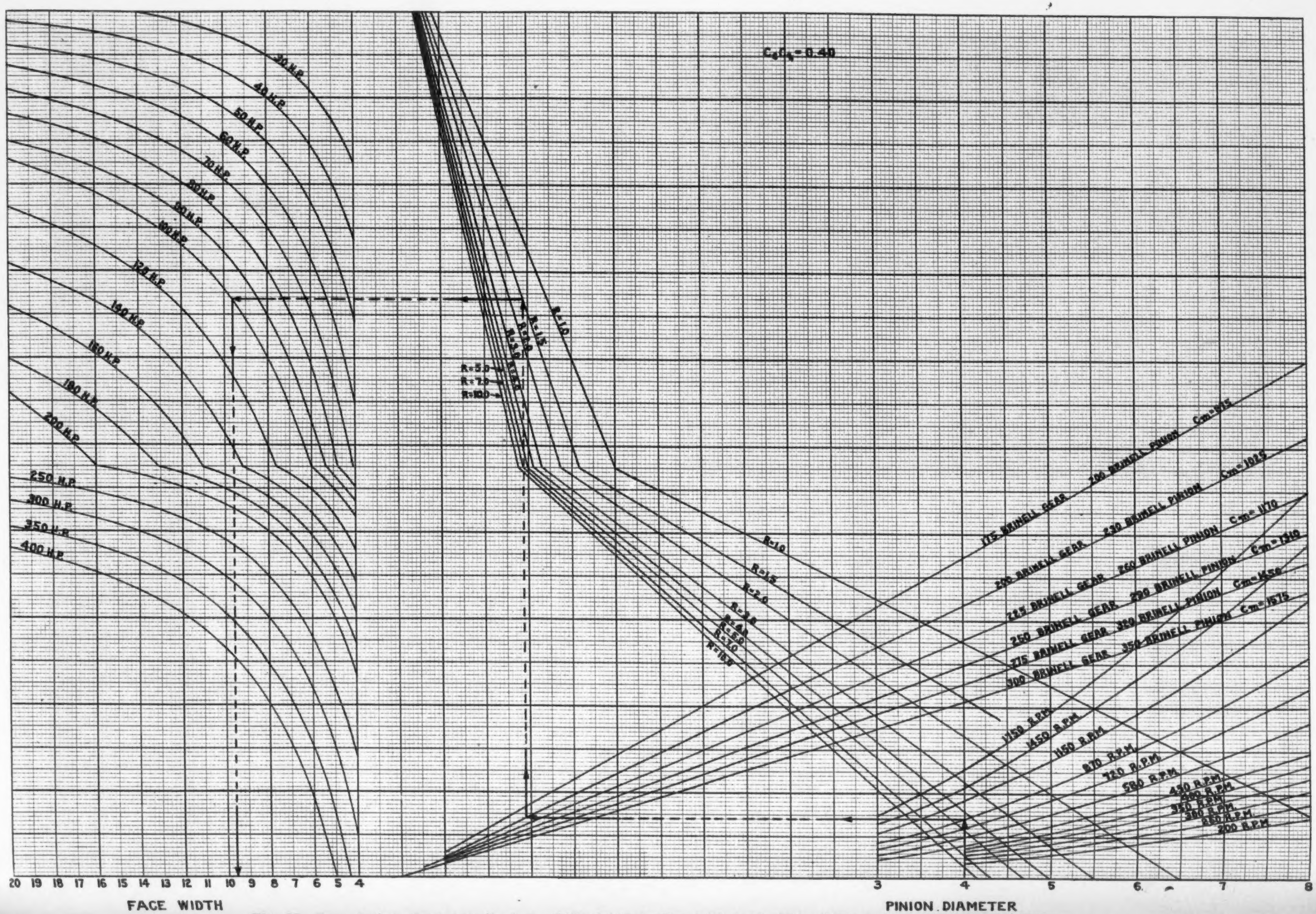

Fig. 36—Gear design chart permits face width determination with loads from thirty to four hundred horsepower

charts have been prepared from which selections can be made in a few minutes. Referring to *Figs.* 35 and 36 it will be noted that by following through from an assumed diameter of pinion to the *RPM*, to the material hardness, to the ratio, then horizontally to the horsepower curve, the vertical from that point will give the required face width. If the ratio of diameter and face width obtained is unsuitable, another trial must be made. Where the face width is specified the diameter required may be quickly obtained by working back through the chart from the other end. These charts can also be used for rating since the intersection of the vertical from the face width, and the horizontal from the diameter, ratio, speed and material, shows the horsepower. Several charts are used for different ranges in order to obtain greater accuracy.

The velocity factor C_v and the inbuilt factor C_i are automatically taken care of when these using charts.

Explains Use of Chart

To illustrate the use of the chart, assume the following problem: Determine the diameter of the pinion and face width of a set of gears to transmit 12 HP at 580 RPM with a reduction ratio of 1.5:1. The pinion to be of .40-.50 carbon steel, 200 brinell; and the gear, .30-.40 carbon steel, 175 brinell.

Using *Fig.* 35, assume a pinion diameter of 2 inches. Starting at 2-inch pinion diameter, trace vertically to the curve for 580 RPM, then horizontally to the intersection of the materials hardness curve for the 175 brinell gear, then vertically to the ratio curve for 1.5:1 ratio, then horizontally to the horsepower curve for 12 HP and finally vertically downward to the face width scale where we obtain a required face width of 6.7 inches.

Checking the ratio of face width to pinion diameter, we find that this exceeds 2.5 and we must necessarily assume a larger pinion diameter.

Using a pinion diameter of 2.6 inches and proceeding as before we obtain a face width of 3.1 inches or approximately 3⅛ inches. This ratio of face width to pinion diameter is satisfactory.

These charts are made up for a C_c C_q value of 0.4 which is quite safe for ordinary proportions. Where greater refinement is necessary the C_c and C_q values should be obtained in the manner previously outlined and a face width correction made by dividing the correct C_c C_q value and multiplying by 0.4.

For other than a uniform load with eight hour service, the equivalent horsepower must first be computed by multiplying the specified horsepower by the factors shown in Table I. Likewise, where other than oil lubrication is used, the equivalent horsepower is found by multiplying the specified horsepower by:

0.8 for compounds	To be applied only where
0.75 for heavy E.P. lubricants	their use is indicated

After the diameters and face width have been determined, the pitch must be selected. This is a matter which requires the application of considerable judgment. For heavy shock loads as in rolling mills, extremely heavy pitches are necessary. For high speed turbine driven reduction gears, fine pitches are essential.

A convenient method of making pitch selections is obtained from an adoption of Day's imaginary straight tooth[7]. This assumes a single tooth carrying all the load and subjected to direct shear. It has the virtue of neglecting both the pressure angle and the helix angle; and thus can be used for more general application than certain modified forms of the Lewis equation.

For ordinary application

$$DP = 0.157 \times FW \times S_s / TP$$

where

DP = diametral pitch

FW = face width

S_s = allowable shear stress (to be taken from *Fig.* 27)

TP = total tangential tooth pressure

Too coarse a pitch is a disadvantage because the lineal contact is decreased. *Fig.* 37 shows a picture taken at the beginning of a test in which two sets of gears were used, one a coarse stub tooth, the other a long tooth with half the circular pitch of the stub tooth. The load was increased in increments until failure of the coarse pitch gear prevented further testing.

The overlap in the axial plane should in no case be less than 25 per cent and preferably 100 per cent or more. Where there is any question or doubt, it should be checked by the rule

$$N_a = (FW \times \tan VH) / CP$$

A number of tests were conducted to check this theory of gear rating. Some of these consisted of continuous operation at rated load, and some at overload. In one test speed reducer gears were operated at low speed, and the load increased in increments to determine the point where shear failure took place. Microphotographs were taken similar to *Fig.* 38, which shows a section of the tooth wherein

[7] "Herringbone Gears," by Percy C. Day, annual meeting, A. S. M. E., 1911.

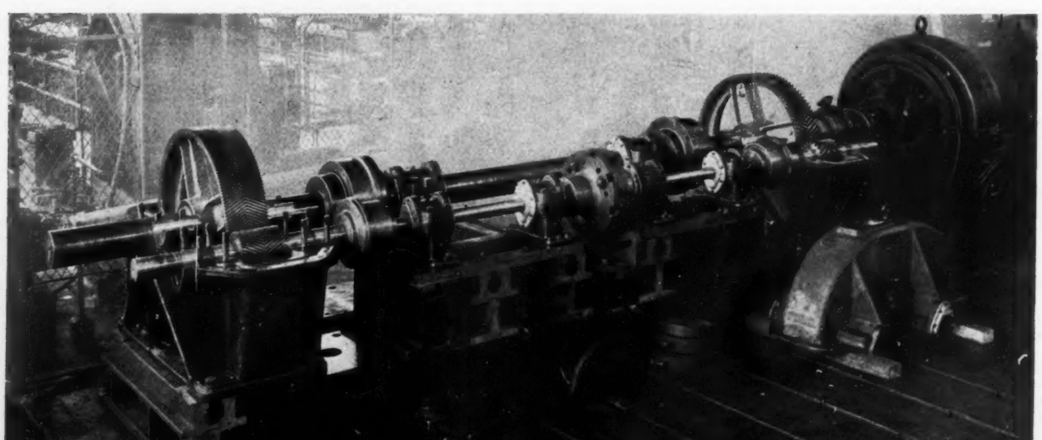

Fig. 37—Gears of different pitches and materials were tested by this testing arrangement

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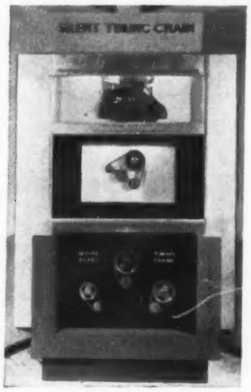
POWER TRANSMISSION NEWS

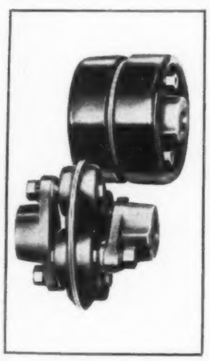
Stroboscope at the World's Fair



To show the slow movement and smooth action of Morse Silent Chains, the stroboscope is used by the Morse Chain Company in the Borg-Warner exhibit in the Travel and Transport Building at the Century of Progress.

New Application of Rubber in Power Transmission

A comparatively new type of flexible coupling is coming into more general use in connecting shafting and is replacing the conventional universal joint where the angular misalignment is not excessive.



The coupling consists of two forged steel hub members mounted on the shafts to be connected. Between these is carried the floating center to which each hub is separately connected through a pair of resilient rubber trunnion blocks. These trunnion blocks, 90° apart, are set under pressure into a two-piece riveted housing of pressed steel. The blocks are molded over and permanently fastened to steel cores, or bushings. These steel cores, in opposite pairs, are rigidly bolted to the end hub members of the coupling.

The especially developed resilient non-cold-flow rubber blocks are responsible for the extreme flexibility of the coupling, providing for greater parallel and angular misalignment than any other coupling. These rubber blocks are all on a common center, equally radial from a common axis, consequently this coupling, at any angle, will run smoothly and true.

This is known as the "Morflex" coupling and is made by the Morse Chain Company, of Ithaca, N. Y. They cite as an example of its efficiency a recent installation, joining a motor and generator. Before this coupling was put on, a semi-universal was in use, which was noisy and apparently absorbed a lot of power, causing the motor to heat badly and causing difficulty with the bearings.

After the "Morflex" coupling was installed, the noise disappeared, the motor ran cool and the owner's coupling troubles were over.

As there is no metal-to-metal in the assembly, there is no rattle or wear, consequently no lubrication is ever required. Furnished in a variety of popular sizes, for from $\frac{3}{8}$" to 3" shaft diameters.

20 Years of Service

By the time any piece of equipment has rendered 20 years of service in any one plant, it may be regarded as an old friend. A prominent knitting mill in Rockford, Ill. (name on request), has 40 such friends in the shape of 40 Morse Silent Chain Drives that have been running for a generation. These are 2½" units, transmitting 13 h.p. between 17 and 49 tooth sprockets.

Chain Drives

UNDER ALL CONDITIONS

MEET every transmission demand. Flexible, no power loss, low maintenance cost, low first cost, long life, chain drives give you definite performance and operating economies. Adaptable, can be applied to centers you want.

Ask Morse engineers for further information on the specific application of chain drives to your transmission needs.

MORSE CHAIN COMPANY

A Division of Borg-Warner Corporation

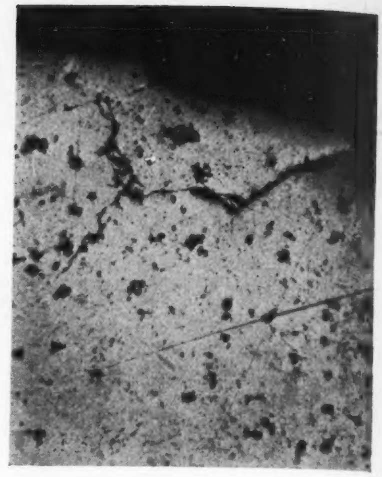
ITHACA, NEW YORK

a shear crack has started under the surface.

The results from a special testing machine for large gears served to verify this shear stress theory further. Small gears traverse every inch of face of the large gear. The load is applied free of any shock, the speed is extremely low and only a very light oil is applied. The pressure per inch of face used for 5 D.P. gears is 4000 pounds. This results in a shear stress of 25,000 pounds per square inch. The shear yield of the gear material is 40,000 pounds per square inch.

The gear proportions obtained by the method outlined are conservative, and allow for small variations from the optimum on the various elements entering into the determinations. The formula provides an equitable means of accounting for the influence of the various design factors on the safe load carrying capacity of helical and herringbone gears. To the manufacturing department must be delegated the responsibility for maintaining the equally important qualitative and quantitative standards as regards heat treatment, tooth spacing, profile runout, lead and accumulated error.

Fig. 38 — The shear stress theory was checked by making microphotograp h s of the contacting teeth after they were subjected to stresses equal to the shear yield point



The author wishes to acknowledge the work of George Maurer who collaborated in the development of this formula, P. C. Day, A. H. Candee, Hyman Ledeen and others for checking and suggestions, and to E. P. Connell, vice president of Falk Corp., who provided the appropriation which made it possible.

Gear Experts Discuss Schmitter's Analysis

Additional comments on this article by authorities in the gear industry appear immediately following the first section of this article published in the June issue of MACHINE DESIGN. *The comments begin on page* 61 *of that issue.*

Best Available Information

I CONSIDER that Mr. Schmitter has brought together the best available present information in regard to the load-carrying capacity of helical and herringbone gears. His work represents an important step toward arriving ultimately at a method of rating gears which may be applicable to all types. The job is by no means finished.

It must still be made clear that the general machine designer should consult the gear manufacturer for the best design of all important gear drives.

—A. H. Candee,
Gleason Works

Valuable to the Gear Industry

THIS paper represents considerable work and is very valuable to the gear industry, a group greatly indebted to Mr. Schmitter and to the Falk Corp. for the analysis. There are a few statements made to which the writer would like to add the experiences of the South Philadelphia Works of the Westinghouse company.

With reference to "pitting" we agree with the author that the cause is not definitely known. A contributing cause is probably the sliding action of the teeth as shown in *Fig. 5B* (M.D. June). On the pinion tooth the sliding is "away" from the pitch line while on the gear teeth the sliding is "toward" the pitch line. Plastic flow results from this tooth action tending to form a "wire edge" on the pinion teeth. On the gear tooth the same action tends to build up metal at the pitch line. This results in a higher loading at the pitch line and a line of pits at the pitch line, as indicated in *Fig. 5A* is the result. We have measured the teeth of a new gear at the pitch line and found that after several months operation the thickness will be increased slightly.

Under "gouging," page 45 of the June issue, the statement is made that "this trouble is not incurred if loads are kept within normal value." We have had this occur in several instances on gears operating at practically no load. This seems to be due to the sharp edge of the entering tooth not necessarily digging into the mating tooth but shearing off the oil and preventing proper lubrication. The remedy is to round the corners or to relieve the tips of the teeth.

On page 53 (M.D. June), it is stated "the practice of loading the second reduction set higher than the first set is practically universal." On account of the duration of contact being greater on the second reduction, the oil film has more time to be punctured. It is our tendency at the present time to reduce the loading on the second reduction gear.

Under "deflection" (page 46 of this issue, the maximum face width is given as 2½ diameters. This we assume is for a two-bearing pinion or for the face width of each helix of a three-bearing pinion.

Reference is made (page 46) to gear proportion to give an integral number of teeth in contact. We have used this on smaller gears and have found it possible to obtain this in a standard gear by proper choice of pitch and helix angle.

The formula given for loading includes factors not usually varied by a partciular manufacturer. We find that for our application we only need know the allowable

load per inch face per inch diameter of the pinion. Judgment is used based on past experience and the application of the gear in determining these facts, after which the gear proportions are easily determined.

—IRA SHORT,
Westinghouse South
Philadelphia Works

Suggests Additional Testing Before General Use

OUR analysis of W. P. Schmitter's article on "Design and Capacity of Helical and Herringbone Gears" leads us to question the advisability of applying the data, as he has submitted it, to the general field of gearing. We are of the opinion that the formulas developed agree fairly well with recognized practice for the class of gears usually encountered in enclosed standard speed reducers employing materials, tooth forms, finishes, etc. which are in common use at the present time. This work is largely a theoretical analysis and a compilation of data which has been obtained from a number of sources and Mr. Schmitter is to be complimented on the manner in which he has correlated the results of a number of investigations along this line. However, in applying this data to the general field of gearing, we are of the opinion that additional testing should be done to determine whether or not

TABLE I

Applications of Data to Specific Cases Examples Considered

Example No.	Center Dist.	Ratio	Pinion R.P.M.	Gear R.P.M.	Pitch Line Velocity	Helix Angle degrees	Pres. Angle degrees	Face Width inches
1	8	1.8:1	571	308	839	20	20	5
2	8	1.8:1	1704	919	2500	20	20	5
3	8	4:1	1000	250	839	20	20	5
4	8	4:1	2980	745	2500	20	20	5
5	8	9:1	2000	222	839	20	20	5
6	8	9:1	5960	662	2500	20	20	5

the data can be applied in its present form with safety to the large general field of gearing where other than "standard speed reducer" practice has been used. A large amount of this kind of testing has been done in connection with straight spur gears by Professor Buckingham and a similar amount of testing will, no doubt, be advisable for helical gears before we can safely apply the data.

We have attempted to analyze the data from this viewpoint and in Table I is shown a tabulation of figures developed when this data is applied to specific cases and compared to other data which is in common use at the present time.

Using the example cases of Table I we calculated the "Permissible Horsepower Capacity" by several formulas in common use at the present time, with the results shown in Table II.

A review of this information indicates the wide variation in permissible loading obtained with present formulas, and illustrates the need for additional testing to determine the proper field of application for each formula or a better formula. Whether or not a "universal" formula applicable to the whole field of gearing can be developed remains an unanswered question.

Fig. 12 (M. D. June, p. 55) tends to be confusing from the text, and an example sketch would materially help the reader to visualize a point Mr. Schmitter is making.

On pages 54 and 59 (M. D. June) the article discusses

TABLE II

Comparison of Rating Formulas

Example No.	Pitch (DP)	Permissible Horsepower Formula A	Formula B	Formula C	Formula D
1	5	173	99	132	54
	25	194	23	32	54
2	5	430	246	215	107
	25	482	58	52	107
3	5	118	83	111	31
	25	134	24	32	31
4	5	293	206	181	61
	25	334	61	53	61
5	5	63	51	68	15
	25	74	22	29	15
6	5	156	128	112	31
	25	183	55	48	31

Formula A is the data covered in Mr. Schmitter's article.
Formula B is the A. G. M. A. recommended practice for herringbone gears.
Formulas C and D are rating formulas used by two of the leading gear manufacturers.

contact length and determination of length of contact lines, and it is interesting to compare the data as submitted with test results as described by E. J. Abbott and F. A. Firestone in their article "Measurements of Instantaneous Tooth Contacts in Spiral Bevel Gearing" in the May, 1934 issue of *Mechanical Engineering*. They have included graphs showing the readings obtained on the individual teeth of the gearset by the electrical method, and a table showing the effect of load and lapping on contact angles. We are of the opinion that more tests of the type conducted by E. J. Abbott and F. A. Firestone should be made before we can safely use the data as developed by Mr. Schmitter.

Figs. 35 and 36 as application charts for the data submitteed in the article, are well arranged and will, no doubt, be quite useful in making trial selection. However, in using these charts or similar charts which might be developed for condtions other than the one which has been used in the illustration, it will be necessary to bear in mind that a trial selection is being made which must of necessity be still further checked to determine the proper "pitch" to be used.

Mr. Schmitter states (page 50 of this issue) that after the diameters and face width have been determined the pitch must be selected, and states that this is a matter

TABLE III

Service Application Factors Considering Type of Prime Mover

Driven Apparatus	Driving Machine: Motor or Turbine	Steam Engine	Gas or Oil Engine
Generators Centrifugal Fans and Blowers Centrifugal Pumps and Compressors Belt and Chain Conveyors Wood Working Machinery	1	1.3	2
Pulp Grinders and Beaters Tube Mills—Cement Kilns Bell and Rod Mills	1.3	1.7	2.5
Reciprocating Pumps Crushers Mine Hoists and Elevators Rotary Positive Pumps and Blowers Ammonia and Air Compressors Oil Well Pumping	2	2.5	3

which requires the application of considerable judgment. Our opinion is that the question of beam strength is of the same relative importance as the question of surface resistance and an analysis similar to that which Mr. Schmitter has made with respect to surface resistance should also be made in regard to beam strength if successful design and application of gears in the general field is to be accomplished.

Should Consider Prime Mover

The table of service application factors as shown (page 44) under the heading "Table I" is based purely on a consideration of the type of driven load. We are of the opinion that it is necessary to also consider the type of prime mover and would suggest a table similar to Table III. (Page 54)

Our analysis of Mr. Schmitter's data points to the fact that he has included in his formula the factors which are usually considered as being the critical factors in determining the permissible loading of a set of gears. However, the data assumes conditions which may or may not exist in particular installations with respect to such items as accuracy of tooth spacing and tooth contours, finish produced on the gear teeth, alignment of shafts and type of lubricant, and it would appear to be desirable to have a table of modifying factors similar to the table of service factors which could be used by the average person in designing or applying gears to the general field. Others who have worked on a formula such as is covered in this article, have included such tables, and we are of the opinion that they will be found useful.

—R. S. Marthens,
Nuttall Works,
Westinghouse Electric & Mfg. Co.

Author's Closure

The comments of the various gear authorities are most interesting and I am well pleased at the general agreement expressed therein. In developing an equation of this type one has to be guided by the present state of the art, and it is quite natural that some of the factors such as materials and velocity will be open to discussion, whereas others which represent the influence of the geometrical relations will be exact.

As Mr. Sykes points out, the values of the factor C_m are debatable especially at higher hardnesses. For instance, the effect of the material on the rating in the proposed equation little more than doubles in going from a hardness of 150 brinell to 350 brinell. It may be that we have been unduly conservative at the higher hardnesses, but the test data we have available would not indicate this.

Action Is Largely Rolling

Mr. Sykes draws attention to the difference between the action of teeth meshing together and the action of rollers bearing on each other. Fortunately, the action is largely rolling at the pitch line where one tooth is generally carrying the load and where the pressure is a maximum, so that we are not very far off when we consider teeth as rolling cylinders.

Conformity between the proposed equation and that of Mr. Sykes is better than that obtained with any other method of rating that has been examined to date, with the possible exception of the British standards.

I quite agree with Professor Buckingham that modifications may be desirable when more concrete data is available as to the influence of tooth errors on the dynamic loads. It would be my guess that when test results of this nature are obtained, the dynamic loads will be of considerably less magnitude than has been found to be the case with spur gears. I am sure that if Professor Buckingham says that the dynamic loads are not directly proportional to the applied load, that such will be found to be the case but, in accepting that statement, we must not lose sight of the fact that there are velocity effects other than an increase in contact stress from tooth errors. There is a pressure velocity function which cannot be entirely ignored. The important question is not what tooth stress we have at some fractional load, but how much additional applied load can be safely carried after allowing for the stresses produced by the tooth error. If the applied load is large compared to the latter, the error obtained because of our assumption will be small.

Shaft Diameters Would Be Larger

The load which a set of gears has to carry may vary from a purely friction load to an inertia load. It would be expected that the magnitude of the driving and the driven masses would bear some relation to the load being carried by the gears. Certainly the shaft diameters would be larger in order to carry the greater loads and these will affect the rigidity of the system approximately as the fourth power of their diameter. To illustrate the benefits of an increase of the elasticity constants, let us examine the effect on test run D shown in the ASME Research Publication "Dynamic Loads on Gear Teeth," page 32. For the 5th reading we find that at 524 feet per minute there is a tooth load of 1296 pounds. If we introduce a torsionally resilient coupling, the impact load would be decreased from 1296 to 994 pounds or 23 per cent. It is my thought that the error obtained when assuming average elastic conditions is as great as that obtained when considering the dynamic load proportional to the applied load, and especially so in the case of helical and herringbone gears.

Since the oil film thickness varies with the load, the capacity to absorb errors at the mesh without affecting the velocity of the masses will at least bear some slight relation to the applied load.

Individual Analyses Not Practical

When test values for the dynamic effects are made available for helical and herringbone gears we must not attempt to generalize the elastic conditions. There is too much variation in the magnitude of the dynamic loads with different types of drives. The constants should not be assumed the same for a rod mill drive as for an agitator. Individual detailed analyses are not practical but an attempt should be made to incorporate the differences in a series of dynamic factors. These might possibly be combined with the service factors and listed alongside specific applications.

I am quite sure that Professor Buckingham's anticipation of excessive rating at lower speeds will not be realized. The most consistent criticism made of the formula by those contemplating its use was that it did not permit as high a loading at low speeds as has been practiced in the past. Such criticism can easily be understood when it is considered that it is possible to operate within the endurance limit of the material and still obtain a good many years service because of the small number of con-

tacts obtained in a given length of time when running at such slow speeds. It is also my opinion that the stress values are suitable for the higher speeds as will be found in applying the equation to a high speed drive. The pressure per inch of face per inch of diameter for an 8 to 1 ratio set of turbine driven gears on a naval vessel operating at a pitch line velocity of 9000 feet per minute, using full depth teeth, 17-degree pressure angle, 30-degree helix angle, would be as follows:

$$\frac{TP}{FW \times PD_p} = C_m \times C_{cq} \times C_r \times C_v \times C_i \times (1/C_s)$$

$$= 875 \times 0.5 \times 0.89 \times 0.45 \times 0.425 \times \frac{1}{0.8} = 93 \text{ pounds at}$$

maximum speed (used only occasionally). This is slightly higher loading than used at the present.

The velocity factor $78/(78 + \sqrt{V})$ reduces the allowable running load a greater amount than does the factor $1200/(1200 + V)$ for all speeds up to 225 feet per minute. At a speed of 500 feet per minute there is only 10 per cent difference in the value of the factors as obtained by the two velocity factors. I am not sure that I understand exactly what is meant by slow speeds.

Conservative Loadings Obtained

The point has been made by both Professor Buckingham and Mr. Holloway that variations in the order of accuracy between several plants will, to some extent, affect the actual load carrying capacity of the gears produced by each. This is quite true, although the differences are not nearly as great as in the case of other types of gearing; partly because of the nature of the engagement with herringbone gears; partly because the equipment for producing them is, on the whole, more accurate; and partly because, as Mr. Holloway points out, the manufacture of helical and double helical gears has been controlled by a few companies. The values applied in the constants of the proposed equation are such that conservative loadings are obtained when accuracy is maintained within reasonable limits. When the errors are excessive they will be reflected in greater noise and shorter life. It is in the interests of simplicity to accept certain prevailing standards of accuracy rather than to set up several arbitrary standards.

With regard to the effect of torsional deflection, a satisfactory ratio of pinion diameter to face width will keep the stress variations within the limits permitted by the factor of safety. It is found that bending does more to concentrate pressures than does torsion.

I note Mr. Short's observation to the effect that his practice is to reduce the loading on the second reduction set. It ought to be pointed out that this practice is limited to high speed double reduction turbine drives. We have investigated the practice of a number of companies as regards speed reducers and slow speed gears and the tendency in every case has been to use a higher loading on the second reduction set. For the present, however, it would appear prudent to limit such practice entirely to gears of the speed reducer class.

I was especially interested in Mr. Marthens' statement that the formula agrees fairly well with recognized practice for speed reducers but that he questions the advisability of applying the data to the general field of gearing. Mr. Marthens has given considerable study to the make-up of the formula for speed reducer use and, no doubt, his opinion as to its correctness for that field of application is based on considerable deliberation. It appears that similar study has not been given to the question of applying the formula for general application and there is some doubt in my mind as to whether he has person-

VALUES OF HORSEPOWER RATING

Example No.	Ratio	R.P.M. Pinion	D.P.	No. of Teeth	Horsepower Rating: Formula "A" Proposed wear formula	Strength check	Formula "B" AGMA modified Lewis formula	Formula "C" Westinghouse formula	Formula "D" Sykes formula
1	1.8:1	571	5	29/52	120*	130	95	132	143
			25	143/257	135	26*	22	32	143
2	1.8:1	1704	5	29/52	302*	388	239	215	353
			25	143/257	340	78*	57	52	353
3	4:1	1000	5	16/64	80*	127	94	111	78
			25	80/320	91	26*	22	32	78
4	4:1	2980	5	16/64	202*	380	235	181	195
			25	80/320	230	77*	54	53	195
5	9:1	2000	6.25	10/90	43*	102	70	73	39
			25	40/360	50	25*	20	29	39
6	9:1	5960	6.25	10/90	110*	305	177	120	97
			25	40/360	126	76*	46	48	97

*Indicates governing rating.

ally reviewed the figures as given in the examples. The calculations shown cover a fairly coarse pitch and an extremely fine pitch without regard to tooth strength or the observation of several well-known design rules. The examples are apparently worked out for a 250 brinell pinion, 225 brinell gear, and full depth teeth. Examples 5 and 6 would require the use of an eight-tooth pinion when using 5 D.P. which is not generally considered practical and especially so in this case because the pinion tooth would be pointed. Furthermore, the pinion diameter would be less than one-third of the face width, which is bad.

The accompanying table shows the values of horsepower rating I obtained for the examples given by Mr. Marthens and they differ somewhat from those obtained by him. In examples 5 and 6 a pitch of 6.25 has been used in place of 5 D.P. The values shown for the Sykes equation are based on a C factor of 1.0 and a material factor 1.6 as indicated on page seven of the *Farrel-Birmingham Gear Book*. Horsepower ratings for the contact stress equa-

tion are based on results obtained with equation (*a*) shown on page 44 of this issue.

Values for the Sykes equation for 25 D.P. should really not be considered because they violate his rules for maximum numbers of teeth.

The results shown in the column headed "strength check" have been obtained by solving the suggested pitch equation shown on page 50 of this issue for allowable horsepower. I do not, for a moment, wish to give the impression that the actual physical strength of the gear teeth is as low as shown in the strength formula because it is used here merely to limit the loads which would be obtained by indiscriminate selection of pitches. I cannot agree with Mr. Marthens in his statement that the question of beam strength is of the same relative importance as the question of surface resistance. The above figures indicate that the variation between the several formulas is not nearly as large as Mr. Marthens has supposed, and it is hoped that further checking will convince him that the formula can be extended beyond the range of speed reducer application.

Consider Magnitude of Stress

Mr. Marthens has called attention to the work of Professor Abbott and Firestone in the measurement of instantaneous tooth contacts. It should be remembered that their investigations have been limited to unloaded gears. For gears of the degree of malleability considered in the proposed equation, a fairly close approach to the ultimate would be obtained after the contacts have had an opportunity to develop during the running-in period. The question should not be limited to the lineal evaluation of the load carrying surfaces, but should consider chiefly the magnitude of the stress as determined from the areas of contact.

Mr. Marthens indicates a desire for more corroborative testing. We have completed a little more than 15,000 hours testing in this connection, all but 2000 of which has been on speed reducers of 3, 3¾, 12, 20 and 34-inch centers. We have a 4¾-inch center testing machine operating continuously.

Considerable space has been devoted in this article to those factors which are new. The presentation has been somewhat academic because of the desirability of having a thorough criticism and review of the individual factors. It is to be expected that they will be rearranged to suit the needs and fancies of the individual. For rating purposes the equation can be reduced to the form

$$H.P. = K_m \times D_s \times F_i \times C_r$$

where K_m is the material and pressure constant obtained for the particular tooth form and material used by a particular manufacturer; D_s equals pinion diameter factor which is equal to

$$PD_p^2 \times C_v \times RPM/126{,}000$$

Curves can be plotted for values of D_s for various pinion diameters and *RPM*. F_i equals the inbuilt face width factor which equals $FW \times C_i$ and can also be plotted in a simple curve. C_r, the ratio factor, can easily be obtained by mental calculation.

Mr. Candee has suggested another simplification as shown in the following abstracts from recent correspondence.

"I regret particularly that you have not seen fit to rearrange your C_q and C_c factors. As you have them, the user is required to look up three values, because C_q is obtained by dividing *LA* by *NBP*. The process of simplification can be carried another step further, thus:

$$C_q\, C_c = \frac{K(LA)}{NBP} \sin VP \cos VP$$

$$= \frac{K\,(LA)}{CP \cos VP \cos VBH} \sin VP \cos VP$$

$$= \frac{K\,(LA)}{CP} \times \frac{\sin VP}{\cos VBH}$$

$$C_c = K\frac{LA}{CP} \text{ and } C_H = \frac{\sin VP}{\cos VBH}$$

"Then the contact factor C_c may be described simply as the number of circular pitches contained in the length of the line of action, which is independent of helix angle; and the other factor C_H is the only one affected by helix angle. This makes only two values to be looked up in place of the three required in the present arrangement.

"Your contact length factor C_q, ratio factor C_r, and pinion diameter all vary with numbers of teeth. It seems more logical to me to take care of all the effects of tooth number in one contact factor. The same can be said of your contact length factor C_q and curvature factor C_c, both of which involve pressure angle and helix angle.

"One reason why I have been interested in giving so much attention to the formula for tooth load in helical gears, is that with the proper modification the same general method of calculation applies to spiral bevel gears, and if the correct practice becomes established in one case it also helps the situation for the other case. Thus, the only change necessary in adapting the general equation to spiral bevel gears is to replace the terms $nN/(n+N)$ by $nN/\sqrt{n^2+N^2}$ and perhaps make some adjustment of the numerical coefficient 0.95 expressing the ratio of the minimum to the average length of contact.

"I shall be glad to hear later whether or not you come to agree with my suggestions."

—W. P. SCHMITTER
Falk Corp.

Articles on this and allied subjects published in previous issues of MACHINE DESIGN include:

"Hypoid Gearing Affords Interesting Possibilities in Design," Sept., 1929, p. 20.

"General Principles of Toothed Gearing," by John V. Martenis, Feb., 1930, p. 31, April, 1930, p. 34, and July, 1930, p. 34.

"Four-Speed Transmission Units Indicative of New Gears Trends," by Austin M. Wolf, April, 1930, p. 27.

"Proposes Unconventional Tooth Form for Non-Metallic Gears," June, 1930, p. 45.

"Designing Elliptical Gearing by Simple Formulas," by F. B. Fuller, Jan., 1931, p. 25.

"Meeting Modern Conditions in New Gearing Formulas," by George H. Middleton, Aug., 1931, p. 41, Sept., 1931, p. 41.

"Stresses for Heat-Treated Gears," by George H. Middleton, Oct., 1931, p. 55.

"Bringing Gearing Formulas Up to Date," by G. M. Van Voorhis, Nov., 1931, p. 47.

"Modern Gear Efficiency Exceeds Limits Used in Most Designs," by W. H. Himes, Feb., 1932, p. 28, April, 1932, p. 42.

"Determining Horsepower of Gears," by P. R. Guirl, March, 1932, p. 51, also Sept., 1932, p. 41.

"Force Components in Spur Gear Train," by Paul R. Guirl, July, 1932, p. 41.

"Optimum Size of Intermediate Gears," by Frederic B. Fuller, Aug., 1932, p. 40.

"Predicting Worm Gear Efficiency Easily and Accurately," by H. J. Watson, Oct., 1932, p. 31.

"Deserting Tradition in Developing Special Drive," by Forrest E. Cardullo, May, 1933, p. 18, June, 1933, p. 37.

"Gear Technology Is Discussed," June, 1933, p. 27.

"Smaller Drive Is Feasible with Differential Gears," by Ray T. Congleton, Sept., 1933, p. 11.

"Utilize Full Efficiency of Involute Gearing," by James A. Hall and Harold S. Sizer, Oct., 1933, p. 20.

"Maintaining Tooth Contact," April, 1934, p. 20.

"Reducing Speed in Limited Space," May, 1934, p. 26.

"Use Governs Choice of Gear Steels," June, 1934, p. 33.

"Determining Capacity of Helical and Herringbone Gears," by W. P. Schmitter, June, 1934, p. 40.

NOTEWORTHY PATENTS

ALLOYS that melt at extremely high temperatures tend to freeze so quickly when the source of heat is removed that it is impossible to handle them in ladles like low melting alloys, or pour them directly from the crucible in which they are melted into stationary molds and secure good castings. A process of casting centrifugally these high-melting alloys such as tungsten carbide compositions, has been embodied in an invention conceived by L. H. Brown for Haynes Stellite Co.

The casting machine, *Fig.* 1, employed in the process, incorporates a furnace 10 having a crucible 11 that holds the constituents of an alloy while they are being fused by an electric arc from an electrode 12. Electric current is connected to conductors 15 and 21. Ingredients of the alloy are fused in about 35 seconds. While this is taking place the sprue of cold mold 13 is brought opposite pouring aperture 20 by swinging the table 22, and electric motor 17 which rotates mold 13 is started.

The casting operation is performed by an operator who grasps the handle 19 and tilts the

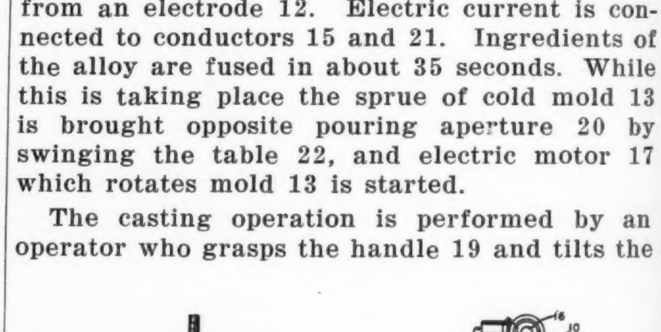


Fig. 1—Machine for casting high-melting alloys employs a rapidly rotating mold. Heating is effected by an electric arc

entire casting machine on its trunnions 14 so that the molten alloy in crucible 11 is poured out through aperture 20 into the rotating mold. When the unit has been tilted about 90 degrees the electric current is disconnected from the electrode. Because the mold is comparatively cold the alloy hardens instantly.

The patent covering this method of casting

alloys is designated No. 1,962,705.

A PROBLEM is solved by a unique invention relating to devices for transmitting the major rotary movements of a shaft subject to hunting or rapid oscillations of relatively small amplitude without introducing any lag in the major transmitted movements. The hunt is especially objectionable where readings of a gyroscopic compass are magnified by coarse and fine repeater compasses. In the past this has been accomplished by a lost motion device which although moderately successful in eliminating the hunt, introduces a lag when the ship is turning so that the transmitted reading is in error during that period. Bruno A. Wittkuhns has invented

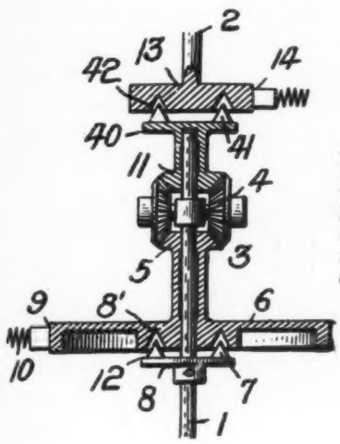


Fig. 2—Device for transmitting the major rotary movements of a shaft subject to hunting or rapid oscillations of relatively small amplitude without introducing any lag in the transmitted movements

a device that eliminates the hunt without introducing lag. Sperry Gyroscope Co. Inc. is assignee of the patent designated No. 1,959,144.

Elements of the device are shown in *Fig.* 2. The driving oscillatory shaft is shown at 1 and the driven shaft at 2. Between the two shafts is differential gearing 3. Planetary arm 4 of the gear train is connected to and turned from shaft 1. Another arm 5 of the train is connected to one portion 6 of a lost motion device 12, connecting 6 to shaft 1.

The lost motion device comprises a pair of cones 7 mounted on collar 8 pinned to shaft 1, and partially entering cone-shaped recesses 8′ in member 6, thereby providing a lost motion between the cones and recesses, which may be adjusted in amount by raising or lowering collar 8. A light friction brake 9 held against the periphery of 6 by spring 10, holds this arm of the differential unless positively turned. The third or driven arm 11 of the train drives driven shaft 2 through a similar lost motion connection.

To explain the theory of operation, it must be assumed that the lost motion or play in the two devices is equal and that the amount of play in the lower lost motion device 12 is about twice that of the normal extent of oscillation of cones 7. As shaft 1 oscillates through this angle, part 6 will not be moved and since it is held by brake 9 the turning of the planetary arm through the same angle will rotate arm 11 through double

that angle. Therefore, if the same amount of play is allowed in the device 40 as in 12, pins 41 will be oscillated merely to touch the walls of the recesses during the normal hunting action but not to move it.

SPRINGS have been utilized by B. M. Leece in a device which permits connection of two aligned shafts so that variations in speed occurring suddenly in one shaft may be transmitted gradually to the other. A patent, No. 1,962,993, for this invention has been assigned to Leece-Neville Co., Cleveland.

In the normal operation of one unit of this type, *Fig.* 3 *A*, the flat sides of springs 16 which are in engagement with the low points of cam 20, form a driving connection that is more or less positive for constant speed. When there is sudden acceleration however, driving ring 12 tends to run ahead of driven member 19. This causes springs 16 to move over the surfaces of the cam toward the high points, flexing the springs. The high spots on the cam must not travel across or beyond the center points of the springs. To prevent such action, stops 17 are provided which engage the springs when the deflection of the latter reaches a safe limit. When the springs are flexed, each of their leaves is individually free to move lengthwise upon adjacent members. Moreover, flexing of the springs shortens the distance between their

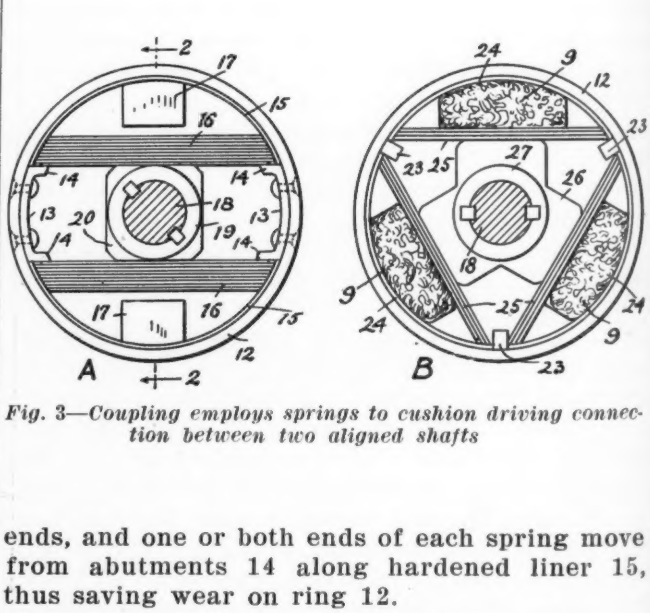


Fig. 3—Coupling employs springs to cushion driving connection between two aligned shafts

ends, and one or both ends of each spring move from abutments 14 along hardened liner 15, thus saving wear on ring 12.

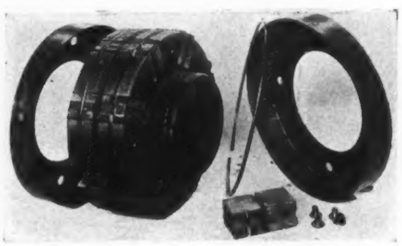
In another variation of the idea, *Fig.* 3 *B*, ring 12 has three abutments spaced equidistantly. Three multiple leaf springs 25 are mounted between liners 24 and the corresponding pairs of abutments 23. Instead of rigid stops for the springs felt pads 9 are employed, which are compressed when the springs flex, therefore tending to limit the flexure. In addition to providing a cushioned stop they serve to dampen any vibratory action of the springs.

New Materials and Parts

Coupling Employs Steel Plates

SIMPLE, positive, yet flexible connections between motor or engine and driven machinery are provided by a new flexible coupling developed by Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. The cushioning action of the coupling, shown herewith, protects both driving and driven equipment against injury from shocks or vibration, and helps to absorb starting or momentary overloads. It also compensates for slight misalignment.

The coupling consists of two identical cast steel flanged halves, two sheet steel cover plates, two snap or retaining rings and from 5 to 22 flexible elements, depending on its size. Machined radially around the periphery of each flange are rectangular slots to receive the cushioning and torsionally flexible elements. The torque transmitting element is a rectangular steel plate, and the cushioning elements are made from high grade hydraulic packing. In assembly, the torque elements are placed in the center of a slot with a cushioning element on each side. Thus the cushioning elements are subjected only to compression under load. The flexible elements and the periphery of the coupling flanges are provided with a groove into which snap rings are inserted, preventing radial or lateral movement of the elements.



Torsionally flexible cushioning elements transmit the torque in improved flexible coupling

Develops New Bronze Alloy

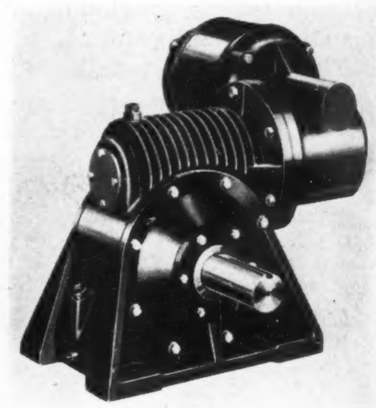
A NICKEL-ALUMINUM bronze said to have extraordinary corrosion resistance and physical properties has been developed by Chase Brass & Copper Co., Waterbury, Conn. The alloy is composed of approximately 92 per cent copper, 4 per cent nickel and 4 per cent aluminum. The metal is being recommended for the stringent requirements of power plants, marine work, oil refineries and the chemical industry. Excellent cold-working properties are said to be

one of its principal characteristics. Its tensile strength ranges from 50,000 to 115,000 pounds per square inch and to as much as 150,000 pounds for cold drawn wire. The alloy is being offered in the form of sheets, rods, wire, tubing and pipe.

Announces New Reduction Units

A NEW series of motorized double worm gear reduction units has been added to the line of such equipment manufactured by Janette Mfg. Co., Chicago. These new units, shown herewith, consist of motors up to one horsepower connected

Motors used with double worm gear reduction units are available in a number of different types



to a train of two worm gear reductions in a wide series of standard available ratios, ranging from 96:1 to as high as 8100:1. Motors for these speed reducers are available in direct current, single phase and polyphase types. Motors are ball bearing, and all gear shafts operate on tapered roller bearings. Worms and gears are designed for high torque applications and consist of hardened and polished steel worms and bronze gears.

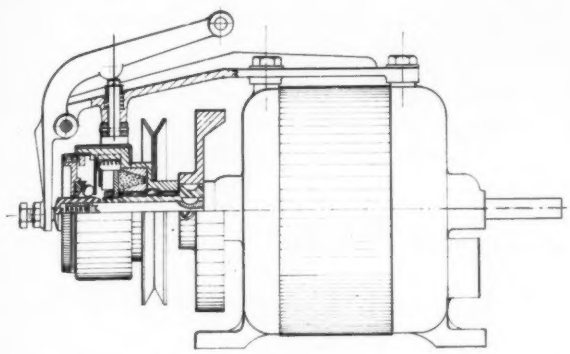
Introduces New Electrode

GENERAL purpose heavily coated electrodes for welding with the shielded arc on mild steel, known as Fleetweld No. 7, have been announced by Lincoln Electric Co., Cleveland. The new electrode is designed particularly for high speed and single pass welding. It has a high burn-off rate and low splatter loss. The electrode's arc characteristics are such that it is particularly suitable for welds where fit-up is poor.

Clutch Unit Incorporates Brake

A BRAKE which is simple yet positive in action and which operates automatically with the clutch control, stopping it almost instantly, has

been incorporated in the design of the Driver clutch manufactured by Walker-Turner Co. Inc., Plainfield, N. J. This clutch, first announced on page 58 of the January, 1934, issue of MACHINE DESIGN, is a relatively small, compact unit, sensitive, positive in action and free from jerky operation. Friction elements are molded asbestos



Zinc die castings are employed in clutch which has molded asbestos friction elements

and cast iron, while a friction area of exceptional size is obtained through the use of a simplified form of double cone. Many parts of the clutch, such as the clutch shell, are of zinc die castings. As it was thought in the design that in some applications in the presence of grit the zinc would not be sufficiently hard, a steel shell has been used as an insert in the die casting. This clutch, shown herewith, has been designed to operate at any speed up to 5000 revolutions per minute, and its action is not effected by centrifugal forces.

Permanently Lubricate Motors

INITIAL lubrication of the type KH washing machine motor brought out by General Electric Co., Schenectady, N. Y., is designed to last for the life of the domestic washing machine on which the motor is installed. The correct amount of lubricant is provided at the factory, and no

Washing machine motors are filled with lubricant at the factory and do not need further reoiling

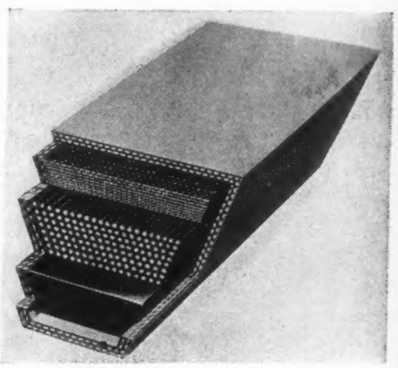
provision is made for reoiling. This new motor, shown herewith, is a resistance split-phase start, induction-run motor. It has constant speed, induction motor running characteristics and a large reserve capacity. It is available in ratings

of 1/6 and ¼ horsepower, 110 and 220 volts, 60, 50 and 25 cycles.

In addition to its lubrication features, all ¼ horsepower motors have the same mounting and overall dimensions, fit into the same space and mount in the same cradle. The motors are permanently fixed in a live rubber mounting which is treated with a compound impervious to oil and is designed to eliminate objectionable noise caused by torque vibration and by endwise and radial out of balance.

V-Belts Are Balanced

PRESERVATION of balance has been particularly designed for in the construction of the new multi-V belts of B. F. Goodrich Rubber Co., Akron, O. In the belts, shown in the accompanying illustration, the three layers are balanced to resist stretch and cover wear. Belts are compounded to reduce internal heat to a minimum.



Three layers in new multi-V belts are balanced to resist stretch and cover wear

Improves Mountings for Bearings

A NEW type of bearing mounting for mine car service which uses a double cup bearing instead of two single cups has been designed by Timken Roller Bearing Co., Canton, O. The shrunk in place steel dust collar which featured previous designs has been retained, but it has been modified so that it provides a labyrinth seal in conjunction with the inner closure.

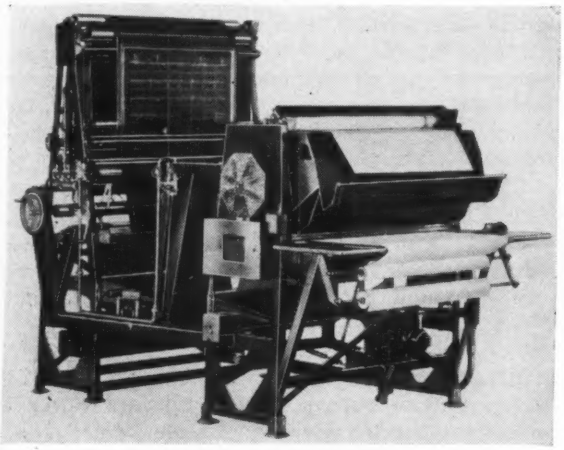
Outer sealing is insured in the new design by the elimination of separate end caps on the wheels, the unit being cast in one piece, thus preventing loss of lubricant in service or entrance of foreign matter into the bearing.

Blueprinter Is Improved

THREE units, a blueprinting machine, a washing unit, and a potashing and drying unit, comprise the new Model 11 blueprinting equip-

ment of C. F. Pease Co., 813 North Franklin street, Chicago. In the equipment, shown herewith, the blueprinting machine can be operated independently by means of a simple clutch, or the machine can be purchased as a separate unit with the other units added later if desired.

The equipment as standard is made in one size only, for paper up to 42 inches wide; however, on special order it can be built for paper up to 54 inches wide. It can be wired to operate on 220 volt direct or alternating current and is powered with a variable speed ¼-horsepower drive motor with a combination gear and sprocket chain drive, all fully enclosed for safety purposes. Three lamps provide a range of printing speed from 4 inches to 12 feet per minute on direct or 60 cycle alternating current. Directly opposite the lamps is an exhaust fan for constantly circulating the air and forcing it from the printer, thereby reducing the temperature of the lamp globes to a minimum and cooling the contact glass.



Speed regulation of blueprinting machine is obtained by a gearshift and rheostat

The machine has a special gearshift underneath the feed table providing for two speeds and neutral. Additional printing speed regulation is obtained by a special hand-operated dial connected by direct shaft to the rheostat which provides instant and accurate change of printing speed of just the proper degree.

Reproducing Machine Is Versatile

BLUEPRINTS, Van Dyke negatives, blue line prints, black and white prints or ozalied prints can be made with the new model E machine of the Shaw Blue Print Machine Co., Newark, N. J. Tracings are fed into the machine on a continuous roll of sensitized paper, or precut sheets may be used. Tracing up to 42 inches width by any length can be printed, and the tracings, after passing around the half cylinder of glass, are returned to the operator.

Extra Copies

MACHINE DESIGN'S June and July issues containing the two sections of W. P. Schmitter's article, "Determining Capacity of Helical and Herringbone Gearing" have been sent to all names on our mailing list.

Additional copies are available on request at the regular price, 35 cents per issue. It will be worth your while to file these copies

FOR REFERENCE

Manufacturers' Publications

ALLOYS (STEEL)—Lukens Steel Co., Coatesville, Pa., is distributing a booklet on Cromansil steel which describes the material completely, gives physical properties and chemical composition and includes pertinent information on methods of fabrication and possible applications of the steel.

BEARINGS—Devoted exclusively to printing press roller bearings, the latest catalog of Bantam Ball Bearing Co., South Bend, Ind., presents several distinctive types of bearings developed to meet special conditions, including a two-row taper roller bearing with wide inner races and a one-piece cup, chrome plated, which on the floating end allows the race to act as a piston in a cylinder, facilitating the sliding necessary to take care of expansion and contraction.

CONTROLS (ELECTRICAL)—Ward Leonard Electric Co., Mount Vernon, N. Y., is distributing catalog inserts on its line of heavy duty relays which have two poles normally open and two poles normally closed, and on its battery charging rheostats and resistors.

CONVEYORS—Platform conveyor chain of the roller type is described in bulletin V-109 of Whitney Mfg. Co., Hartford, Conn., which gives engineering details on the chain and the sprockets used in connection with it.

COUPLINGS—Flexoid industrial couplings are described in a leaflet of Smith Power Transmission Co., Cleveland, which gives available sizes and ratings.

DESIGN DEPARTMENT—Special purpose photographic printers are the subject of a new bulletin brought out by Paragon-Revolute Corp., Rochester, N. Y. The printers, the primary use of which is to reproduce pencil tracings on tracing cloth, will also reproduce, "same size" any one-sided copy that is semitranslucent in the same manner as a camera with reflected light, through a lens.

DRIVES—Link-Belt Co., Chicago, has issued a new 32-page catalog, No. 1415, covering single, double and triple reduction units of herringbone gear types, and a full line of flexible couplings. The book, profusely illustrated, includes horsepower and other engineering data, dimension diagrams, etc. It may be obtained by application to the company on business letterhead.

FASTENINGS—Woodruff type keys, their advantages, uses and sizes are presented in bulletin V-119 of Whitney Mfg. Co., Hartford, Conn. The bulletin contains complete dimensions for the keys and for the keyseat, key above shaft and keyway.

GEARS—Catalog No. 50, recently brought out by Boston Gear Works Inc., North Quincy, Mass., includes complete information, sizes, ratings, dimensions, etc., on a large number of different types of gears in several materials. Also included is data on speed reducers and chain drives.

MOTORS—An outline of the developments in motor manu-

facture made by Fairbanks, Morse & Co., Chicago, and a description of the line of motors manufactured by the company is given in a recent booklet entitled "More Than a Century of Progress." The booklet similarly discusses pumps and other equipment included in the parts being manufactured.

PUMPS—Worthington Pump & Machinery Corp., Harrison, N. J., is distributing bulletin W-321-B2 on its line of single stage monobloc centrifugal pumps. Rating tables, dimensions, method of selection and typical applications are given.

SCREEN CLOTH—Bulletin No. 89 of Robins Conveying Belt Co., New York, includes detailed information on the company's line of screen cloth of several types.

WELDED PARTS AND EQUIPMENT—The progress of bronze welding is presented in a recent booklet of Linde Air Products Co., New York. The booklet gives procedures for work on cast iron, malleable iron, steel and wrought iron, copper, and brass and bronze.

Research Publications

Thermal Expansion of Bearing Bronzes. An investigation on the linear thermal expansion of bearing bronzes (copper-tin and copper-tin-lead alloys) undertaken at the bureau of standards indicates that the expansion curves in the first heating of cast bronzes are in most cases above the observations obtained in the first cooling. Coefficients of expansion obtained in the second heating are generally higher than those obtained in the first. The addition of tin to copper, or lead to copper-tin-alloys increases the coefficients of expansion. With the aid of charts included in the publication it is possible to predict the coefficients of expansion of other cast copper-tin and copper-tin-lead alloys. Published as RP 665 by bureau of standards. Available through Superintendent of Documents, Government Printing Office, Washington.

The Strength of Screw Threads Under Repeated Tension, by Herbert F. Moore and Proctor E. Henwood. Bolts and studs in structural and machine parts are commonly subjected to axial tension in the threaded portion. It is customary to compute the tensile stress in the threaded portion by dividing the axial load by the area of the cross section at the root of the threads. The result is the average stress on the cross section; but, at the root of the threads, there exists localized stress much higher than this average. It is a matter of common experience that, under repeated loading, fractures occur in service in steel bolts and studs subjected to average stresses at the root of the thread as low as 20,000 pounds per square inch. Stress concentration factors found in this investigation were somewhat lower than those found in previous investigations by the photoelastic method, using models made of pyralin, with polarized light. As a result of the stress concentration factors noted in the present investigation, it is suggested that a safe estimate of the stress at the root of the thread would be, not the nominal value of P/A, but $3P/A$ for ordinary structural steel, and $4P/A$ for heat treated alloy steels commonly carried in stock, where P is the load in pounds and A the area at the root of the thread in square inches. Published as bulletin No. 264 by Engineering Experiment station, University of Illinois, Urbana, Illinois.

INDEX TO ADVERTISERS

*Advertisements appear in previous issues.

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MACHINE DESIGN is a monthly technical publication conceived, edited and directed expressly for those executives and engineers responsible for the creation and improvement of machines built for sale, and for the selection of the materials and parts to be used.

Business Announcements and Sales Briefs

LOUIS ALLIS CO., Milwaukee, has opened a direct factory branch office in the Bona Allen building, Atlanta, Ga. The territory from this office, in charge of George C. Gardner, includes the states of North Carolina, South Carolina, Georgia, Florida, Alabama, Tennessee and Mississippi.

* * *

Louis C. Lustenberger, of Carnegie Steel Co. sales staff at Pittsburgh, has been appointed assistant to the vice president and general manager of sales, assigned to special sales duties.

* * *

Charles W. Dean has been made district manager of the Cleveland office of Crocker-Wheeler Electric Mfg. Co., Ampere, N. J., succeeding the late Fred C. Eckworth.

* * *

Duraloy Co., South Twenty-sixth street, Pittsburgh, has appointed L. M. Utilities Co., Tulsa, Okla., its sales representative in Oklahoma and Kansas.

* * *

Edward C. Gainsborg has been appointed general manager of Roller Bearing Co. of America, Trenton, N. J. He formerly was associated with the company for several years as sales manager.

* * *

D. A. Nemser recently was appointed development engineer for International Nickel Co. Inc., with headquarters at Hartford, Conn. He was for ten years associated with the metallurgical staff of Pratt & Whitney Mfg. Co.

* * *

Climax Molybdenum Co., 295 Madison avenue, New York, has opened two additional branch offices, one at 310 South Michigan avenue Chicago, in charge of E. R. Young, and another in the First National Bank building, Canton, O., in charge of Paul M. Snyder.

* * *

Bantam Ball Bearing Co., South Bend, Ind., has appointed F. D. Clark as district sales engineer in northwestern Ohio and southeastern Michigan. Offices will be in the Spitzer building, Toledo, O. Automotive bearings of the company will also be handled by G. A. Ashton Co., St. Paul, and Robert Cannon, Kansas City, Mo.

* * *

S. C. Du Tot has been appointed Chicago district manager of sales for Electro Metallurgical Sales Corp., New York. Offices will be in the Union Carbide & Carbon Corp. building in that city, where until recently the Electro Metallurgical corporation has maintained only service offices. J. F. Donovan has been appointed Cleveland district sales manager, a newly created office with headquarters in the Union Trust building, Cleveland. Both men were formerly identified with the New York general sales office of the corporation.

* * *

K. D. Crawford has been appointed Pittsburgh manager of the Taylor-Wharton Iron & Steel Co., High Bridge, N. J., succeeding the late J. R. Smith. Mr. Crawford formerly was associated with the American Frog & Switch Co.

* * *

Frederick B. Heitkamp, formerly sales manager, Cincinnati Milling Machine & Cincinnati Grinders Inc., has resigned to accept a similar position with Lyon Metal Products Inc., Aurora, Ill.

* * *

Republic Steel Corp., Youngstown, O., has appointed Pidgeon-Thomas Iron Co., South Main and Iowa streets, Memphis, Tenn., distributor of its Enduro stainless steel.

* * *

James A. Long, former general manager of Woodward Iron Co., Birmingham, Ala., has been named district sales manager of Macwhyte Co., Kenosha, Wis., maker of wire rope, to succeed the late James A. Boope. Headquarters will be in Birmingham and territory will include Alabama, Virginia, Georgia, Mississippi, Arkansas, Tennessee and Florida.

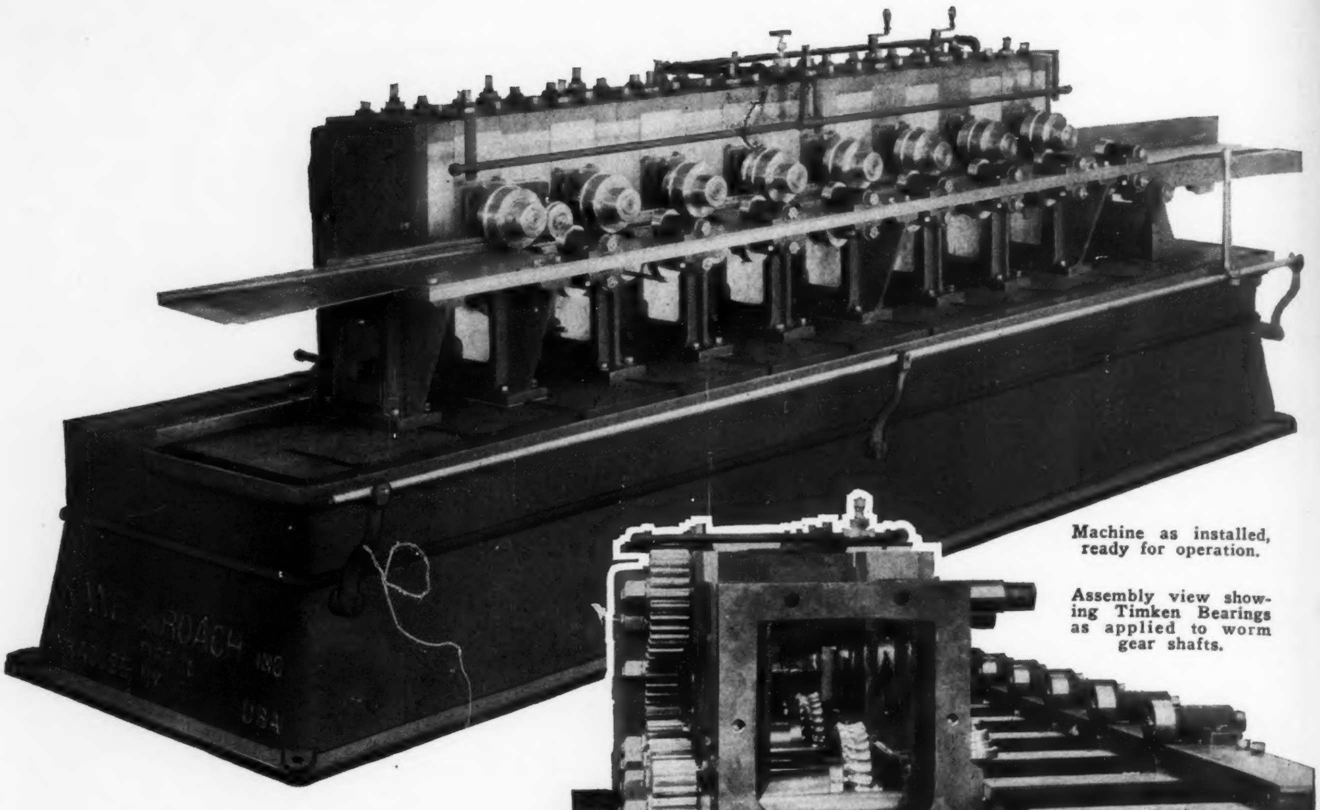
* * *

W. R. Judson has been appointed manager of the Salt Lake City, Utah, district office of the Allis-Chalmers Mfg. Co., Milwaukee. For the past three years he has been managing director of Allis-Chalmers (France), representing the company in Europe. Prior to that time, for 13 years, he was manager of the company's office at Santiago, Chile.

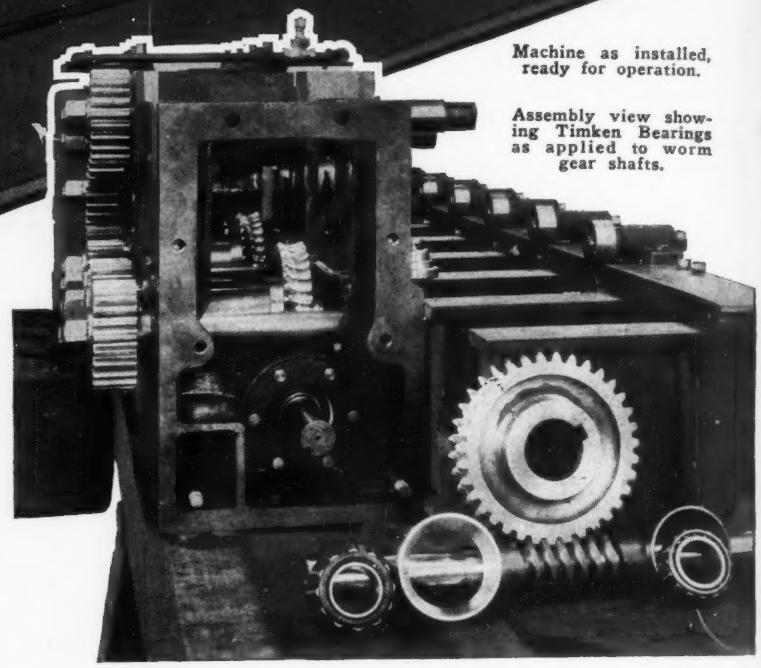
H. E. Weiss, whom Mr. Judson succeeds, has been transferred to the company's office at Kansas City, Mo.

* * *

J. R. Bateman is now handling specialized sales and engineering service of products of Bunting Brass & Bronze Co., Toledo, O., with headquarters in Chicago and territory including Illinois, Wisconsin, Minnesota, Missouri, Iowa and Kansas. J. F. Roberts has been appointed manager of the Bunting Branch office at Dallas, Tex. J. H. Roberts has been named manager of the Kansas City, Mo., branch. F. W. Roberts is the new manager of the metropolitan New York branch.



Machine as installed, ready for operation.



Assembly view showing Timken Bearings as applied to worm gear shafts.

64 TIMKEN BEARINGS

for SPEED, ACCURACY, ECONOMY and LONG LIFE

THIS Kane & Roach Overhung Type Series L Cold Roll Forming Machine equipped with 64 Timken Tapered Roller Bearings is installed at the Autostroy-Doskino plant in Russia.

It is used for hemming automobile running boards. The material is hot-rolled steel, No. 16 U.S.S. gauge, .058" - .056". Running boards are rolled in two lengths—29 35/64", tapered from 13½" to 11⅝" wide, and 39 7/32" tapering from 13¾" to 11¼" wide. Production speed, 100 feet per minute. Machine speed, 1,000 R.P.M. Power input, 15 H.P.

Anti-friction bearing requirements in this equipment are broad and exacting. Continuous and severe radial-thrust loads must be carried; shafts, gears and forming rolls must be held in correct and constant alignment; even roll pressures must be maintained.

Timken Tapered Roller Bearings are able to meet every requirement completely and fully by virtue of the exclusive combination of Timken tapered construction, Timken positively aligned rolls and Timken alloy steel.

That's why Kane & Roach, Inc., use them for these tough cold roll forming machine jobs. That's why you need them in machines of any type. They reduce operating costs, extend machine life and cut maintenance expense to the bone.

THE TIMKEN ROLLER BEARING COMPANY, CANTON, OHIO

TIMKEN *Tapered Roller* **BEARINGS**